<u>İSTANBUL TECHNICAL UNIVERSITY ★ INSTITUTE OF SCIENCE AND TECHNOLOGY</u>

COGENERATION OF ELECTRICITY AND COOLING BY GAS TURBINES

Ph.D. Thesis by

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Department: **Mechanical Engineering**

Programme: **Energy**

AUGUST 2006

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PREFACE

The successive energy crises have stimulated the study of more efficient ways for the use of the available energy in fuels. As consequence new technical plants have been conceived seeking the primary energy conservation. Cogeneration maybe defined as the simultaneous production of electrical or mechanical energy and useful thermal energy from a single energy source. After the process the waste heat can be converted to useful refrigeration by using a heat operated refrigeration system. The use of heat operated refrigeration system help to reduce problems related to global warming, such as the so called green house effect from CO_2 emissions from the combustion of fuels in utility power plant. The absorption systems are more prominent for the zero ozone layer depletion.

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ABBREVIATIONS

AARS	: Ammonia absorption refrigeration system
APH	: Air preheater
ARU	: Ammonia refrigeration unit
C	: Compressor
CE	: Cooling effect
CFC_s	: Chlorofluorocarbon refrigerants
CHP	: Combined heat and power
COP	: Coefficent of performance
CRF	: Capital recovery factor
CV	: Control volume
HRSG	: Heat recovery steam generator
I	: Investment
LBWAS	: Lithium bromide water absorption system
LHV	: Lower heat value
NG	: Natural gas
OGT	: Optimal generator temperature
ST	: Steam turbine
TE	: Exhaust temperature
TR	: Refrigeration temperature
VCRS	: Vapour compression refrigeration system

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LIST OF SYMBOLS

$\mathbf{B_f}$	: Exergy content of fuel input
$\mathbf{B_p}$	: Exergy content of process heat produced
$\dot{\mathbf{C}}$	: Cost rate associated with exergy rate transfer ($/s)
$\mathbf{c}$	: Cost per unit exergy ($/kJ)
$\mathbf{c_p}$	: Constant pressure specific heat (kJ/kg-K)
$\mathbf{c_v}$	: Constant volume specific heat (kJ/kg-K)
$\dot{\mathbf{E}}$	: Exergy rate (kW)
$\dot{\mathbf{E}}_\mathbf{D}$	: Exergy destruction rate (kW)
$\mathbf{h}$	: Specific enthalpy (kJ/kg)
$\mathbf{n}$	: Economic life of the investment
$\dot{\mathbf{n}}$	: Molar flow rate (kmol/sec)
$\dot{\mathbf{Q}}_\mathbf{H}$	: Heat rejected to atmosphere (kW)
$\dot{\mathbf{Q}}_\mathbf{p}$	: Process heat (kW)
$\dot{\mathbf{Q}}_\mathbf{R}$	: Cooling effect (kW)
$\mathbf{R_{PH}}$	: Power to heat ratio
$\dot{\mathbf{S}}_\mathbf{gen}$	: Entropy generation (kJ/kmol–K)
$\mathbf{T_0}$	: Temperature of the environment
$\dot{\mathbf{W}}$	: Power
$\mathbf{y}$	: Coefficient of carbon dioxide
$\mathbf{1-y}$	: Coefficient of carbon monoxide
$\dot{\mathbf{Z}}$	: Cost rate associated with a plant component or a system

Greek letters

$\boldsymbol{\eta}$	: Cycle thermal efficicency
$\boldsymbol{\eta_f}$	: Fuel unitization efficiency
$\boldsymbol{\eta_{II}}$	: Second law efficiency
$\boldsymbol{\eta_{st}}$	: Isentropic turbine efficiency
$\boldsymbol{\eta_{sc}}$	: Isentropic compressor efficiency
$\in$	: Effectiveness
$\boldsymbol{\lambda}$	: Fuel–air ratio on a mole basis
$\boldsymbol{\rho}$	: density (kg/m^3)

Subscripts

a	: Air
ap	: Air preheater
c	: Air compressor
cc	: Combustion chamber
g	: Saturated vapour
gt	: Gas turbine
$\mathbf{n_v}$	: K mol of water vapour
OM	: Operation and maintenance
ph	: Preheater
th	: thermal

Superscripts

CH	: Chemical
CI	: Capital investment
HX	: Solution heat exchanger
KN	: Kinetic
PH	: Physical

COGENERATION OF ELECTRICITY AND COOLING BY GAS TURBINES

SUMMARY

The object of this thesis is to do the thermoeconomic analysis of the gas turbine cogeneration systems where the exhaust gases are used for refrigeration purposes. The thermoeconomic analysis involves thermodynamic considerations as well as the calculation of economic feasibility of such systems and cost rates of the products.

Cogeneration is defined as the simultaneous production of power and heat. In essence it aims to utilize the exhaust heat of prime movers such as gas turbines, steam turbines and gas motors for producing electricity. Thus a more effective utilization of fuel is achieved. This has two important consequences. First of all use of lesser amounts of fuel in context of decreasing fossil supplies and secondly reduced carbon dioxide emissions in view of the global warming concerns. The fact that the exhaust heat may be used in absorption chillers introduces a new direction for cogeneration. Thus besides electricity and process heat, cooling effect may be produced by cogeneration. This application is sometimes called trigeneration in the literature. There are two types of absorption refrigeration cycles that are widely used in practice. These are the aqua–ammonia cycle and the lithium bromide–water cycle. The former can be used for refrigeration at temperatures below 0°C. The latter is generally used in air conditioning systems and the minimum temperature is limited to approximately 4°C.

A numerical model of a cogeneration system consisting of a gas turbine system, heat recovery steam generator, a steam turbine, a pump and an absorption refrigeration unit was formed in this study. The steam turbine and the absorption refrigeration unit are coupled to the gas turbine system through the heat recovery steam generator. The gas and steam cycles were considered as steady flow systems, air and the combustion products were assumed to be ideal gas mixtures. Natural gas (methane) was used as fuel. Two programs were written to realize the computations of the model.

The first program does the first law analysis of the system, calculates the mass flow rates of fuel and air, temperatures, pressures and exergy rates at all points of the system.The second program calculates the cost rates and cost per unit exergy at all state points of the system. The numerical model was simulated for different values of the pressure ratio of the compressor, cost of the natural gas, the investment cost of the gas turbine and the investment cost of the steam turbine. Furthermore an economic analysis was done to compute the payback period of the system for different parameters.

It was found that the cost of electricity that can be produced by such a system, would vary between 0.04 and 0.06 $/kWh, and the cost of the cooling effect would vary between 0.018 and 0.026 $/kWh. These values compare favorably with the current costs of these commodities in the market. The fuel utilization effectiveness has been

found as 70 %, as compared to 50% for the separate production of products. The payback period was found to be between 7 and 9 years.

GAZ TÜRBİNLERİ İLE BİLEŞİK ELEKTRİK ÜRETİMİ VE SOĞUTMA

ÖZET

Bu çalışmanın amacı atık gazların soğutma elde etmek için kullanıldığı gaz türbinli bileşik ısı-güç (kojenerasyon) sistemlerinin termoekonomik çözümlemesidir. Termoekonomik çözümleme, termodinamik çözümlemenin yanında bu tür sistemlerin ekonomik olurluluğunu ve ürünlerin maliyetlerini irdeler.

Bileşik ısı-güç üretimi elektrik ve ısının aynı santraldan elde edilmesi anlamına gelmektedir. Bileşik ısı-güç üretimi temelde, elektrik üretiminde kullanılan gaz türbini, buhar türbini ve gaz motorları gibi ısı makinalarının atık ısısından yararlanmayı amaçlar. Böylece yakıt enerjisi daha etkin kullanılmış olur. Bunun iki önemli sonucu vardır. İlk olarak giderek tükenen fosil yakıtlardan tasarruf etmek, ikinci olarak küresel ısınma kaygısını atmosfere daha az karbon dioksit atarak azaltmak.

Atık gazların abzorpsiyonlu soğutucularda kullanılarak soğutma elde edilmesi bileşik ısı-güç üretimi için yeni bir yön göstermektedir. Böylece elektrik ve proses ısısı yanında, bileşik ısı-güç üretimiyle soğutma etkisi de elde edilebilir. Bu uygulamaya kaynaklarda ‘trijenerasyon’ adı verilmektedir. Uygulamada yaygın olarak kullanılan iki abzorpsiyonlu soğutma çevrimi vardır. Bunlar amonyak-su ve su-lityum bromür çevrimleridir. Birinci çevrim 0 ^{o}C’ nin altındaki sıcaklıklar için kullanılabilir. İkinci çevrim ise daha çok iklimlendirme sistemlerinde kulanılmaktadır ve elde edilebilecek en düşük sıcaklık yaklaşık 4 ^{o}C ile sınırlıdır.

Bu tezde gaz türbini, atık ısı kazanı, buhar türbini ve abzorpsiyonlu soğutucudan oluşan bir bileşik ısı-güç sisteminin sayısal bir modeli oluşturulmuştur. Buhar çevrimi ve abzorpsiyonlu soğutucu, gaz türbini çevrimine atık ısı kazanı ile bağlanmışlardır. Bileşik ısı güç sistemi sürekli akışlı bir sistem olarak alınmış, hava ve yanma sonu gazları mükemmel gaz karışımları varsayılmışlardır. Yakıt olarak doğal gaz (metan) kullanılmıştır.

Modelin hesaplamalarını yapmak için Fortran dilinde iki program yazılmıştır. Birinci program sistemin birinci yasa çözümlemesini yapmakta, yakıt ve hava debilerini hesaplamakta, sistemin her noktasında sıcaklık, basınç ve ekserji akılarını bulmaktadır. İkinci program sistemin her kütle akısı için maliyet akılarını ve birim ekserji maliyetlerini hesaplamaktadır. Sayısal model, karar parametrelerinin değişik değerleri için çalıştırılmıştır. Bu parametreler, gaz türbini çevriminin basınç oranı, doğal gaz fiyatı, gaz türbini ve buhar türbininin maliyetleridir. Ayrıca parametrelerin değişik değerleri için sistemin geri ödeme süresini hesaplayacak ekonomik çözümlemeler yapılmıştır.

Sonuçlar böyle bir sistemden elde edilecek elektriğin fiyatının 0.04 and 0.06 $/kWh, soğutma etkisinin maliyetinin ise 0.018 and 0.026 $/kWh arasında olacağını göstermiştir. Bu değerler piyasada bugün karşılaşılan değerlerin altındadır. Enerjiden yararlanma oranı %70 olarak bulunmuştur. Ayrı ayrı üretim durumunda bu değer %50 olmaktadır. Geri ödeme süreleri 7-9 yıl arasında bulunmuştur.

1. INTRODUCTION

The object of this thesis was to do the thermoeconomic analysis of the gas turbine cogeneration systems where the exhaust gases are used for refrigeration purposes. The thermoeconomic analysis involves the thermodynamic considerations as well as the calculation of the economic feasibility of such systems and cost rates of the products. It is hoped that this study will lead to energy conservation in hot countries where electric power generation and refrigeration are needed simultaneously.

Cogeneration is defined as the simultaneous production of power and heat. In essence it aims to utilize the exhaust heat of prime movers such as gas turbines, steam plants and gas motors used for producing electricity. Thus a more effective utilization of fuel is achieved. This has two important consequences. First of all use of lesser amounts of fuel in the context of decreasing fossil supplies and secondly reduced carbon dioxide emissions in view of the global warming concerns.

The fact that the exhaust heat may be used in absorption chillers introduces a new direction for cogeneration. Thus besides electricity and process heat, cooling effect may be produced by cogeneration. This application is sometimes called trigeneration in the literature.

Cogeneration was used in Europe and especially in former eastern block countries mainly in counjunction with district heating. But it has also gained wide usage in industry around the world in the last 20 years. There are many applications of cogeneration in industrial plants where electricity and process heat are produced simultaneously. These plants in general pay themselves back within 3 to 4 years by savings in fuel.

This thesis consists of five chapters. After the introduction the second chapter is a literature review on cogeneration and absorption refrigeration.

The third chapter discusses the underlying concepts of the model. First of all cogeneration is examined in depth, parameters characterizing cogeneration are explained. A special emphasis is given to gas turbine cogeneration. Secondly absorption refrigeration is considered. Aqua – ammonia and lithium – bromide water systems are explained with the help of two numerical examples. Finally the thermoeconomic principles are examined. The cost balance equation is stated, the formation of cost rate is explained.

The fourth chapter is a detailed explanation of the model. The thermodynamic and economic rules governing the behaviour of each component of the system are examined. The assumptions made in the analysis are given, magnitudes of the parameters of the system are stated.

The fifth chapter is a detailed explanation of the results and discussion. Exergy rates, cost rates and cost per unit exergy were calculated for all state points (streams) of the system. Exergy destruction, relative cost difference and exergoeconomic factor were calculated for all components. Furthermore an economic analysis was done to determine the pay back period of the system for various values of the decision variables.

For the compressor ratio of 10 and 10 MW power production the cost per unit exergy of the cooling effect is 0.1153 $/kWh. The cost per unit energy of the cooling effect is 0.022 $/kWh. The cost per unit energy of the cooling effect in the literature is 0.0256 $/kWh. The cost per unit exergy of the gas turbine electricity is 0.0413 $/kWh. The cost per unit exergy for the steam turbine electricity is 0.083 $/kWh. The industrial cost of electricity in Europe is 0.095 $/kWh.

The pay back period for different parameters including the pressure ratio, price of the natural gas, investment cost of the gas turbine system, the absorption refrigeration system and the steam turbine was found to be between 7 and 9 years. The value of the pay back period in Europe is 12 years.

2. LITERATURE REVIEW

2.1. Introduction

Cogeneration involves the production of both thermal energy, generally in the form of steam or process heat and electricity. The thermodynamic and engineering performance of combustion gas turbine cogeneration systems can be found in the literature (Rice, 1987). The use of process heat to power an ammonia-water absorption refrigeration (AAR) plant is viable and under certain circumstances an economical option. While lithium Bromide chillers are becoming more wide spread and therefore their production is standardized to particular need, AAR is an old refrigeration technology, but until recent times it was applied mainly in large scale process plants, mostly in petrochemical industry. New developments in AAR technology in the smaller range appeared in the literature in the last few years and new installations are known Bassols et al. (2003), (Apte, 1999). The study aimed primarily the analysis of application of cogeneration in hot climates where electricity and cooling are simultaneously required.

2.2. Literature Review

Bilgen (2000) has investigated the exergetic and engineering aspects of gas turbine based cogeneration plants. The exergy analysis is based on the first and second laws of thermodynamics. The engineering analysis is based on both the methodology of levelized cost and the pay back period. To simulate these systems, an algorithm was developed. Two cogeneration cycles, one consisting of a gas turbine and the other of a gas turbine and steam turbine to produce electricity and process heat were analyzed. The aim of Bilgen's study was to complement previous studies using exergy concept, to present a modular technique for engineering economics and to develop an algorithm useful for modeling cogeneration systems. The thermodynamic models were based on the methodologies using the first and second laws of thermodynamics and the exergy concept. The engineering methodology was based on standard engineering methodologies for design, cost evaluation and economics of the electrical energy produced and the pay back period of the additional investment for process heat production. For 22 MWe gas turbine cycle the total cost was 7.741 M$, typical product cost without cogeneration was 0.037$/kWh, typical product cost with cogeneration was 0.021 $/kWh, and the pay back period was 0.175 years. While for the gas turbine and steam turbine cycle, the total cost was 9.623 M$, typical product cost with cogeneration was 0.023 $/kWh and pay back period was 0.906 years. In this thesis the investment cost is in the range of 600 to 700 $/kW for the gas turbine system, 1000 to 1200 $/kW for the steam turbine system. The final product is the cooling effect while in Bilgen's study the final product is the process heat.

El-sayed (1992) found that heat and power integration in industries can save both fuel and cost and this is observed in the cogeneration system considered in this thesis. El-sayed found that heat pump assisted cogeneration is one way of integration when the heat / power ratio for a given product is large. It has the advantage of more fuel saving than the conventional grid cogeneration (selling back electricity). With the current state of art of vapour compression heat pumps, the advantage is also economic in many of the situations where power needs do not exceed 30MW, temperature levels do not exceed 67°C and electricity fuel price ratios do not exceed 3. For wider applicability with economic superiority new directions of developments are needed for power driven heat pumps. El-sayed concluded that a power driven absorption heat pump may be the answer.

Colonna and Gabrielli (2003) proposed that the increase in fuel prices and the ecological implications are giving an impulse to energy technologies that better exploit the primary energy sources and integrated production of utilities should be considered when designing a new production plant. The number of so called trigeneration systems installations (electric generator and absorption refrigeration plant) were increasing. This system is adopted in this thesis. If low temperature refrigeration is needed (from 0 to - 40°C) ammonia – water absorption refrigeration plants can be coupled to internal combustion engines or turbo – generators. A thermodynamic study of trigeneration configurations using a commercial software integrated with specially designed modules was presented. The study analyzed and compared heat recovery from the prime mover at different temperature levels. In the last section a simplified economic assessment that took into account prices in different European countries compared conventional electric energy supply from the grid with an optimized trigeneration plant. For a generator temperature lower than the optimal generator temperature, which implies decreasing evaporator pressure, increases the amount of heat flow that can be transmitted to AAR cycle, therefore the generating temperature which maximizes the refrigerating heat flow is 120°C. This corresponds to a heat recovery steam generator evaporating pressure of 0.27 MPa. In this condition the trigeneration system produces 10.14 MW_e, 25.8 t/h of steam 16.2 MW_{th} from which 9.57 MW_{ref} of refrigerating effect can be generated. The energy flow entering the system is 32.84 MW_{th}. The cost per unit energy of cooling effect was found to be 0.0256 $/kWh while in this thesis the cost per unit energy is 0.022 $/kWh .

Rice (1987) has established a heat balance for evaluating various open cycle gas turbines and heat recovery systems based on the first law of thermodynamics. This relates to this thesis as it takes into consideration the gas turbines and recovery systems. A useful graphic solution is presented that can be readily applied to various gas turbine cogeneration configurations. An analysis of seven commercially available gas turbines is made showing the effect of pressure ratio, exhaust temperature, intercooling, regeneration and turbine rotor inlet temperature in regard to power output, heat recovery and overall cycle efficiency. The method presented can be readily programmed in a computer, for any given gaseous or liquid fuel, to yield accurate evaluations.

Huang (1990) discussed the thermodynamic performance of selected combustion gas turbine cogeneration systems based on first law as well as second law analysis. The effect of the pinch point used in the design of heat recovery steam generator, and pressure of process steam on fuel utilization efficiency, power to heat ratio, and second law efficiency, are examined. Results of three systems using state of the art industrial gas turbines show clearly that performance evaluation based on first law efficiency alone is inadequate. A more meaningful evaluation must include second law analysis. The object of this thesis was to do the thermoeconomic analysis of the gas turbine, which involves the thermodynamic considerations. The first program in this thesis does the first law analysis of the system.

Bassols et al. (2002) have shown that in the food industry cogeneration plants are widely used. Many industries use cogeneration plants with either gas engines or turbines to cover their steam, hot water and electrical demands. The combination of absorption refrigeration with a cogeneration plant allows the use of generated heat for the production of cooling effect. Absorption refrigeration plants working with ammonia as refrigerant can be driven either by steam, pressurised hot water or directly with exhaust gases. Examples of typical plants are illustrated in different sectors of the food industry. In this thesis a gas turbine system is used to cover the steam demand. The absorption refrigeration system is coupled to the gas turbine through the steam turbine and the heat recovery steam generator to produce the necessary cooling effect.

Srikhirin et al. (2001) have conducted a literature review on absorption refrigeration technology. A number of research options such as varios types of absorption refrigeration systems, research on working fluids and improvement of absorption processes were discussed. The COP of a single stage ammonia refrigeration system was taken as 0.6. In this thesis a single stage ammonia – water refrigeration system is used. The average COP of the absorption refrigeration system in this thesis has been taken as 0.6.

Siddiqui (1997) has investigated the economic analysis of absorption system components with the aim to optimize the various operating parameters. The absorber, condenser, generator, rectifier, precooler and preheater have been designed using standard procedures and their costs have been estimated based upon material used, fabrication, installation and overhead charges. Four types of refrigerant – absorbent combinations (H_2O – LiBr, NH_3 – H_2O, NH_3 – NaSCN and NH_3 – $LiNO_3$) using either solar collectors, biogas or liquified petroleum gas as the source of heat have been selected. In this thesis the investment cost data for all components are taken as input data for the first program.

Mone et al. (2001) have investigated combined heat and power (CHP) systems which often use absorption technology to supply heating and cooling to a facility. With the availability of gas turbines spanning an increasingly wide range of capacities, it is becoming more and more attractive to utilize CHP via a combination of gas turbines and absorption chillers. They investigated the economic feasibility of implementing such CHP systems with existing commercially available gas turbines and single, double and triple stage absorption chillers. The maximum amount of thermal energy available for the chiller was calculated based on the size of the turbine, exhaust flow rate and exhaust temperature, yielding approximately 300,000 kW of cooling (85,379 tons of refrigeration) for 600 MW power turbine. The annual demand and avoided costs for varying turbine and absorption system sizes were discussed as well, showing that a CHP system is capable of large savings. In this thesis the system study focuses on the comparison of plant configuration for a 3,5,10,15,20 and 30 MW trigeneration system for industrial applications. The cooling effect for 30 MW power turbine is 15823.08 kW (4495.19 tons of refrigeration) .

Adewusi and Zubair (2004) applied the second law of thermodynamics to study the performance of single stage and two stage ammonia-water absorption refrigeration systems (ARS) when some input parameters are varied. The entropy generation (Sgen) of each component and the total entropy generation of all the system components as well as the coefficient of performance (COP) of the ARS were calculated from thermodynamic properties of the working fluids at various operating conditions. The results show that the two stage system has a lower entropy generation and a higher COP while the single stage has a higher entropy generation and a lower COP. In this thesis the first law of thermodynamics, calculates the mass flow rates of fuel and air, temperatures, pressures and exergy rates at all state points. A single stage ammonia refrigeration system is considered.

Misra et al. (2002) have reported that the optimization of thermal systems is generally based on thermodynamic analysis. However the systems so optimized often are not viable due to economic constraints. The theory of exergetic cost is a thermoeconomic optimization technique, combines the thermodynamic analysis with that of economic constraints to obtain an optimum configuration of a thermal system. This technique is applied to optimize a LiBr / H_2O vapour absorption refrigeration system run by pressurized hot water for air – conditioning applications. The mathematical and numerical optimization of thermal systems is not always possible due to plant complexities. Hence a simplified cost minimization methodology, based on “Theory of Exergetic cost”, is applied to evaluate the economic costs of all the internal flows and products of the system under consideration. Once these costs are determined, an approximately optimum design configuration can be obtained. In this thesis the second program calculates the cost rate per unit exergy for all state points of the system. Input data to this program are the capital cost of components, fuel cost and exergy rates at all state points of the system. The input data is generated in the first program.

Usta and Ileri (1999) have discussed the importance of economic optimization of large capacity or industrial refrigeration systems and present the results and conclusions obtained by a computer software which was developed specially to determine the economic optimum values of the design parameters of refrigeration systems. Both liquid chillers and group of cold storage rooms operating at various levels of low temperatures are considered. Various case studies and sensitivity analyses were performed to provide specific numerical examples and to determine the effects of certain parameters. It was found that condenser type, ambient temperature, yearly operating hours, electricity price, real interest rate and refrigerant are the most important parameters in the economic optimum design of refrigeration systems. The condenser temperature for chillers with either water or air cooled condensers were investigated. The optimized condenser temperature is lower up to several degrees, when the yearly operating time is high or the relative interest is low. This is so no matter whether the condenser is cooled by water or ambient air. The condenser temperatures are significantly lower about 33°C for air cooled condenser and 50°C for water cooled condenser. It was found that the systems with lower capacities requires slightly lower condenser temperature. In this thesis two computer programs were written to calculate mass flow rates of fuel and air, temperatures, pressures and exergy rates, the cost rates and cost per unit exergy at all states points of the system.

Kuak et al. (2003) have done the exergetic and thermodynamic analyses of a 500MW combined cycle plant. Mass and energy conservation laws were applied to each component of the system. Quantitative balances of the exergy and exergetic cost for each component and for the whole system was carefully considered. The exergoeconomic model, which represented the productive structure of the system considered, was used to visualize the cost formation process and the productive interaction between components. A computer program was developed which can determine the production costs of power plants, such as gas and steam turbines plants and gas turbine cogeneration plants. The program can be also used to study plant characteristics, namely thermodynamic performance and sensitivity to changes in process or component design variables. In this thesis the second program calculates the cost of the gas, steam turbines electricity, steam from the heat recovery steam generator and the cooling effect from the absorption refrigeration system.

Guarinello et al. (2000) have investigated application of thermoeconomic concepts to a projected steam injected gas turbine cogeneration system, which aims at providing the thermal and electrical demands of an industrial district. The power plant is evaluated on the basis of the first and second laws of thermodynamics. A thermoeconomic analysis using the theory of exergetic cost, was performed in order to determine the production cost of electricity and steam. In this thesis the second program is used to determine the cost per unit exergy of electricity, steam and the cooling effect.

Sun (1997) compiled up to date thermodynamic properties for LiBr / H_2O and H_2O/NH_3 solutions and used them in cycle simulation. Detailed thermodynamic design data and optimum design maps were presented. These results form a source of reference for developing new cycles and searching for new absorbent / refrigerant pairs. They can also be used in selecting operating conditions for existing systems and achieving automatic control for maintaining optimum operation of the systems. In this thesis the thermodynamic calculations related to the aqua – ammonia cycle and the lithium bromide – water cycle are explained by two numerical examples. The methodology follows that given by (Therlkeld, 1970).

White and Oneil (1995) found that the aqua – ammonia cycle is particularly suitable for applications in the process industries, where the refrigerant is required to be at temperatures below 0°C. A modification of the conventional cycle configuration is proposed and investigated. In conventional absorption refrigeration cycles, which employ a volatile absorbent (water), a fraction of the absorbent is carried over into the refrigerant stream. The absorbent is concentrated in the liquid phase in the evaporator and must be removed otherwise this lowers the quantity of useful refrigeration, resulting in a decrease in the thermodynamic performance of the cycle. The contamination in the refrigerant is removed by blowdown to the absorber. The modified cycle employs liquid blowdown from the evaporator to provide reflux for distillation – column generator. This modification can be employed to eliminate the use of fresh refrigerant, from the condenser, as reflux in the conventional aqua - ammonia absorption refrigeration cycle. Simulation of the modified cycle, using the process^{TM} computer simulation package, predicts an improvement in the coefficient of performance (COP) of approximately 5% coupled with a net reduction in total heat transfer area required. In this thesis an aqua – ammonia cycle is coupled to the gas turbine system through the steam turbine and the heat recovery steam generator to produce a cooling effect and suitable for application in the food, pharmaceutical and ice production industries.

Ziegler and Trepp (1984) developed a new correlation of equilibrium properties of ammonia – water mixtures for use in the design and testing of absorption units and especially for heat pumps. The temperature range has been extended to 500°K and the pressure range to 5MPa. The equation of state used is based on those of Schulz. Values of specific volume, vapour pressure, enthalpies and equilibrium constants for mixtures are compared with the best experimental data. The results are presented in the form of vapour pressure and enthalpy – concentration diagrams. In this thesis the enthalpy - concentration diagrams were used to calculate the states and mass flow rates at all nodes of the system.

The COP of the single stage absorption cycle was found as 0.6 while that of the double stage cycle was 0.96. Several types of multi-stage absorption cycle were analysed such as the triple stage absorption cycle and quadruple stage absorption cycle.However an improvement of COP is not directly linked to the increment number of stage. It must be noted that, when the number of stages increase, COP of each stage will not be as high as that of a single stage system. Moreover the higher number of stage leads to more system complexity and increase in cost. Therefore the double stage cycle having COP of 0.96 is the one that is available commercially.

3. UNDERLYING CONCEPTS OF THE MODEL

3.1. Cogeneration

3.1.1. How cogeneration is done

Cogeneration is defined as the production of both electricity and useful thermal energy (steam or process heat) in one operation, thereby utilizing fuel more effectively than if the desired products were produced seperately. The heart of the cogeneration system is a prime mover with waste heat at a high temperature, this requirement may be realized by using different types of prime movers, such as gas turbines, steam turbines, gas engines or combined cycles.

The general concept of a cogeneration system is shown in Figure 3.1

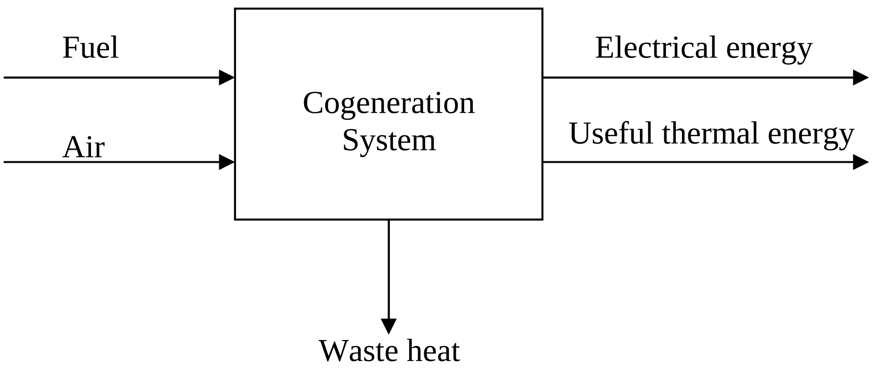

Figure 3.1: General concept of a cogeneration system (Huang, 1990)

3.1.2. Parameters characterizing cogeneration

The useful products of a cogeneration system are electrical energy ($\dot{W}$) and thermal energy or process heat ($\dot{Q}_p$).

One parameter used to assess the thermodynamic performance of such a system is the fuel utilization efficiency (η_f) which is the ratio of all the energy in the useful products ($\dot{W}$ and $\dot{Q}_p$) to the energy of fuel input ($\dot{E}_f$). By definition

$$\eta_f = (\dot{W} + \dot{Q}_p) / \dot{E}_f \qquad \textbf{(3.1)}$$

Since electrical power is worth more than three times the proces heat, the cost effectiveness of a cogeneration system is directly related to the electrical power it can produce for a given amount of process heat. Consequently another parameter commonly used to assess the thermodynamic performance of a cogeneration system is the power to heat ratio. By definition, the power to heat ratio (R_{PH}) is:

$$R_{PH} = \dot{W} / \dot{Q}_p \qquad \textbf{(3.2)}$$

In both the fuel utilization efficiency and the power to heat ratio, power and process heat are treated as equal. This reflects the first law of thermodynamics, which is concerned with energy quantity and not energy quality. But electrical power is much more valuable than process heat according to the second law of thermodynamics. Exergy, the key parameter in second law analysis, is something that is always consumed or destroyed in any real process. A process is better thermodynamically if less exergy is destroyed. Consequently the ratio of the amount of exergy in the products to the amount of exergy supplied is a more accurate measure of the thermodynamic performance of a system. By definition

$$\eta_{II} = (\dot{W} + \dot{B}_p) / \dot{B}_f \qquad \textbf{(3.3)}$$

where

$\dot{B}_p$ is the exergy content of process heat produced and $\dot{B}_f$ is the exergy content of fuel input. η_{II} is the second law efficiency of the cogeneration system.

Efficiencies of different cogeneration systems are compared in Table 3.1

Table 3.1: Comparison of the efficiencies of different cogeneration systems Kartchenko et al. (1998), (Bilgen, 2000)

System	First law efficiency	Utilization efficiency	Second law efficiency
Gas turbine based cogeneration	41.28	86.3	50.06
Steam turbine based cogeneration	26.7	85.1	-
Gas engine based cogeneration	38.1	87.6	-
Combined cycle based cogeneration	-	64.49	49.22

3.1.3. Gas turbine based cogeneration systems

There are many gas turbines in the market today ranging from 1 MW to 100 MW providing a variety of power output, cycle efficiencies, cycle pressure ratios, firing temperatures, exhaust temperatures and exhaust flow rates. Heat recovery of one form or another plays an important part in equipment selection.

A gas turbine based cogeneration system consists of a gas turbine (compressor, combustion chamber and expander) and a heat recovery system for steam production. Steam produced can be used either for process heat or to produce more electric power by a steam turbine. These two cases are illustrated in Figure 3.2 and Figure3.3

3.2. Absorption Refrigeration

The thermal energy produced in a cogeneration system can be converted to a useful refrigeration effect by using an absorption refrigeration cycle. There are two types of absorption refrigeration cycles that are widely used in practice. These are the aqua – ammonia cycle and the lithium bromide – water cycle. The former can be used for refrigeration at temperatures below 0°C. The latter is generally used in air conditioning systems and the minimum temperature is limited to approximately 4°C. The thermodynamic calculations related to these cycles are explained with the help of two numerical examples below. The methodology follows that given by (Threlkeld, 1970).

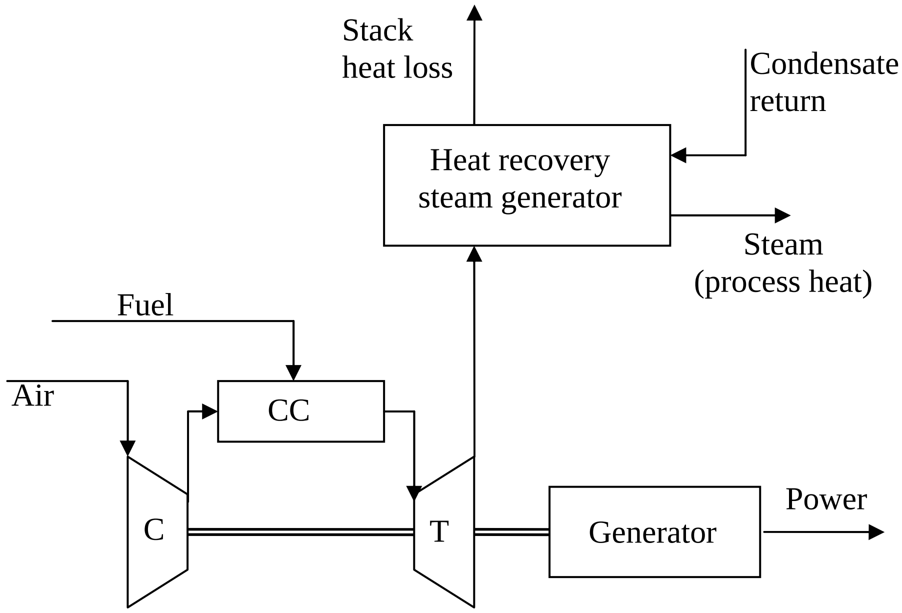

Figure 3.2: Schematic of the cycle for gas turbine electric power production – process heat production

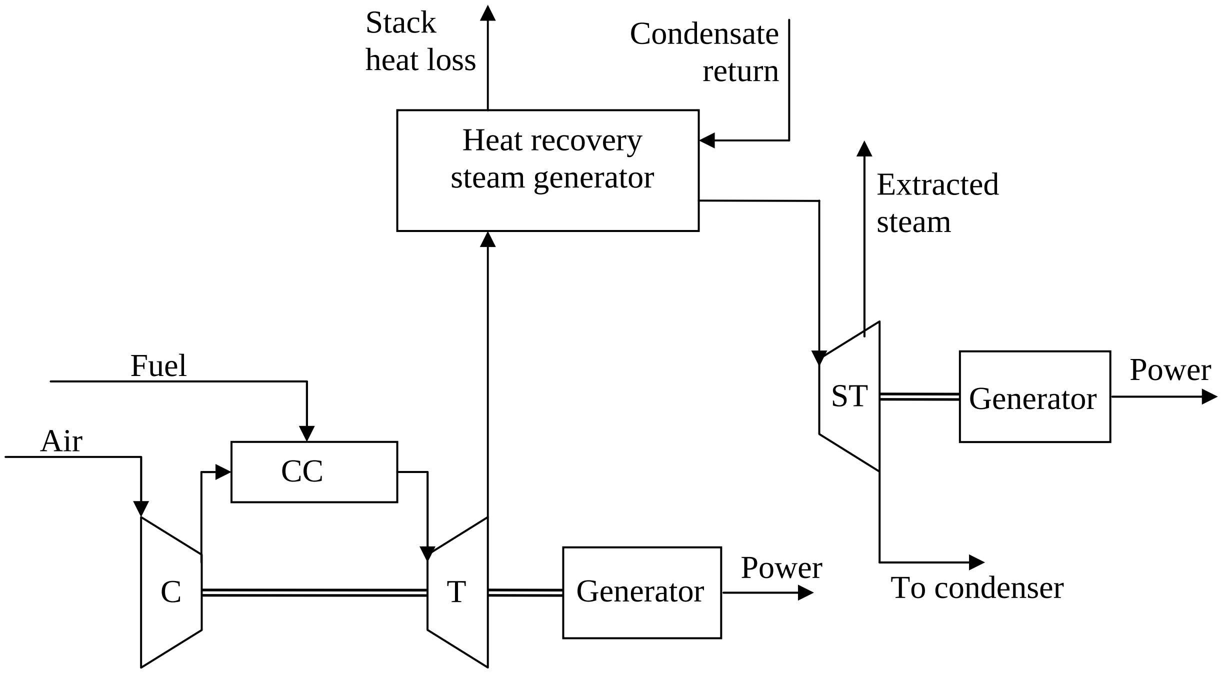

Figure 3.3: Schematic of the cycle for gas turbine electric power production electric power production by steam turbine – process heat production

3.2.1. Ammonia water (aqua – ammonia) absorption refrigeration cycle

The aqua ammonia absorption is one of the oldest methods of refrigeration. Ammonia is the refrigerant and water is the absorbent. An industrial aqua – ammonia absorption refrigeration system is shown in Figure 3.4 .

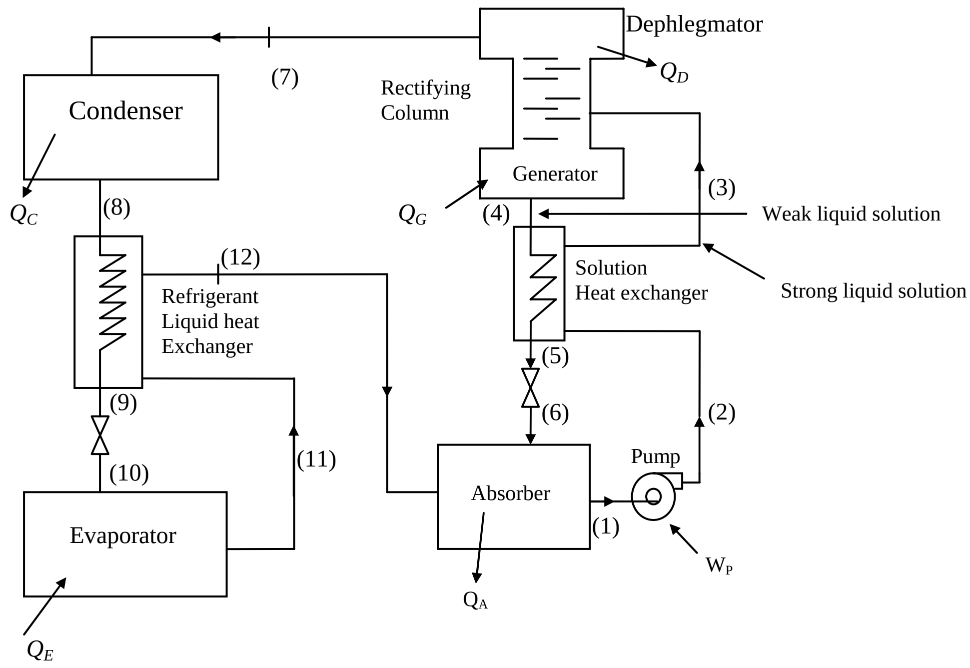

Figure 3.4: An industrial aqua – ammonia absorption refrigeration system (Threlkeld, 1970)

Almost pure refrigerant flows through the condenser and the evaporator. The vapour leaving the evaporator is mixed with a weak liquid solution in the absorber resulting in a liquid solution stronger in the refrigerant. The pressure of liquid solution is then raised to the generator pressure by a pump. By addition of heat in the generator, refrigerant vapour is driven out of the solution. This rather complex process which is partly mechanical partly thermal is realized in the rectifying column. A heat exchanger is placed in the solution circuit between the generator and absorber to improve the performance of the cycle. The generator and condenser are on the high pressure side of the system, while the evaporator and absorber are on the low pressure side Figure 3.4 . Another heat exchanger may be placed between the condenser and the evaporator for the same purpose. A typical aqua - ammonia absorption refrigeration cycle is described below. The evaporator pressure is 0.2MPa and the condencer pressure is 1.5MPa. The generator temperature is 127 °C, temperature of the strong solution is 107°C and the temperature of the vapour leaving the dephlegmator is 87°C.

If the components of the system are considered as steady state steady flow devices and the conservation of energy and mass principles are applied, states and mass flow rates at all nodes of the system can be calculated. Figure 3.5 and Table 3.2 show the results of such an analysis for a 350 kW (100 tons of refrigeration) system. Details can be found in Threlked (1970) and (Derbentli, 2002). The coefficient of performance (COP) of this system was calculated as 0.5 COP depends on the generator and evaporator temperetarures. The average COP of the absorption refrigeration system in this thesis has been taken as 0.6.

Table 3.2: Properties at state points of the aqua – ammonia refrigeration cycle

State	P (MPa)	T (°C / K)	Conc. (x)	h (kJ/kg)	m (kg / s)
1	0.2	32 / 305	0.32	- 50	2.3712
2	1.5		0.32	- 48.4	2.3712
3	1.5	107 / 380	0.32	314	2.3712
4	1.5	127 / 400	0.22	440	2.0672
5	1.5	37 / 310	0.22	22.7	2.0672
6	1.5	37 / 310	0.22	22.7	2.0672
7	1.5	67 / 340	1.0	1390.	0.304
8	1.5	29 / 310	1.0	200.	0.304
9	1.5	31 / 304	1.0	150.	0.304
10	0.2	- 13 / 260	1.0	150.	0.304
11	0.2		1.0		0.304
12	0.2	7 / 280	1.0	1350.	0.304

3.2.2. Lithium bromide – water absorbtion system

In recent years the lithium bromide – water system has become prominent in refrigeration for air conditioning. Water is the refrigerant, lithium bromide is the absorbent. The outstanding feature of the system is the non – volatility of lithium bromide. No rectifying equipment is required, since water vapour can be easily vaporized from the mixture. In comparison with the aqua – ammonia system, the lithium bromide – water system is simple and operates with a higher coefficient of performance. Its primary disadvantage is its limitation to relatively high evaporating temperatures as the refrigerant is water.

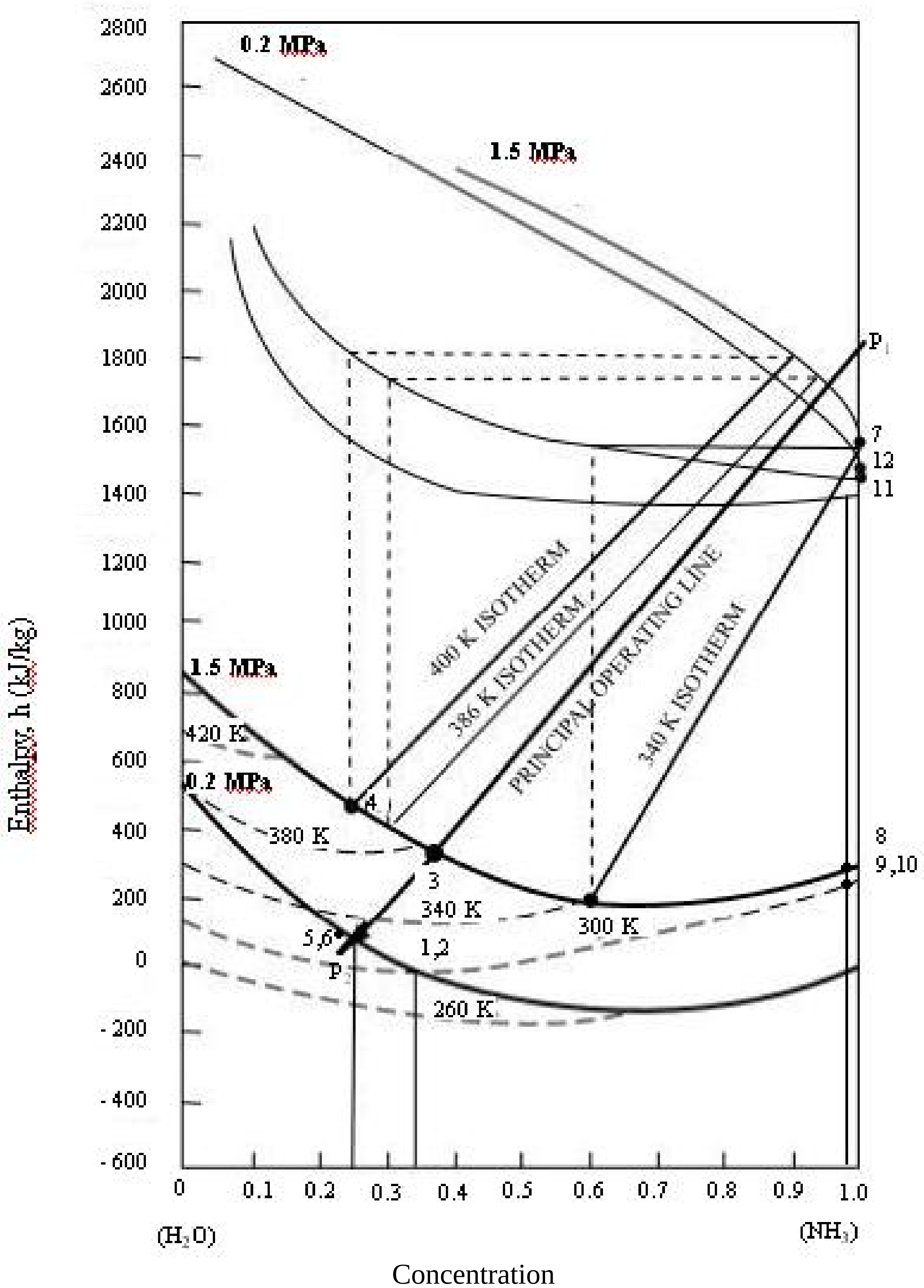

Figure 3.5: Constructions done on the h-x diagram for the aqua – ammonia cycle, Derbentli (2002), (Threlkeld, 1970)

A simple absorption refrigeration system is shown Figure 3.6 . A typical lithium bromide –water absorption refrigeration cycle is described below. The evaporator pressure is 8kPa and the condenser pressure 65 kPa . Note that the system operates under vacuum. The generator temperature is 93°C and the strong solution enters the generator at 82°C.

If the components of the system are considered as steady state steady flow devices and the conservation of energy and mass principles are applied, states and mass flow rates at all nodes of the system can be calculated. Figure 3.7 and Table 3.3 show the results of such an analysis for a 3.5 kW (1 ton of refrigeration) system. Details can be found in Derbentli (2002) and (Threlkeld, 1970). The COP of this system was calculated as 0.78 . The average COP of the absorption refrigeration system in this thesis has been taken as 0.6.

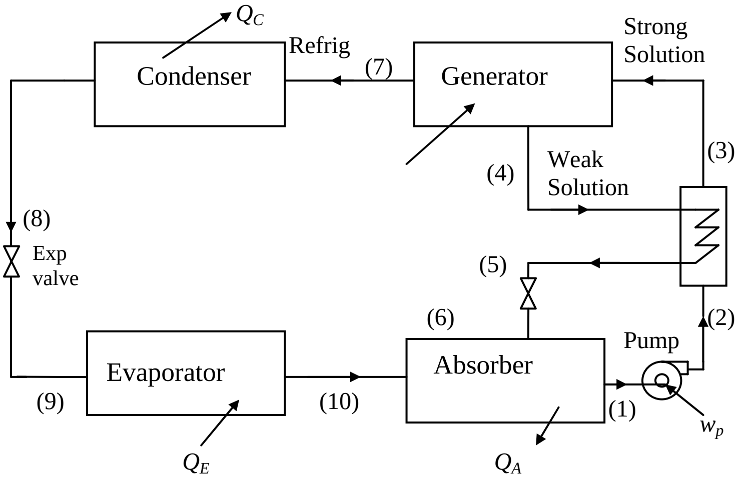

Figure 3.6: A typical lithium bromide - water absorption refrigeration system

Table 3.3: Thermodynamic properties and flow rates for a typical lithium bromide – water absorption refrigeration cycle

State – Point	Pressure p (kPa)	Temperature T (°C)	Concentration x	Enthalpy h (kJ kg)	Flow Rate m (kg/s)
1	8	38	0.60	....	0.02
2	65	….	0.60	....	0.02
3	65	82	0.60	- 81	0.02
4	65	93	0.65	- 63	0.018
5	65	....	0.65	....	0.018
6	8	....	0.65	....	0.018
7	65	93	0.00	2677	0.0015
8	65	38	0.00	158	0.0015
9	8	5	0.00	158	0.0015
10	8	5	0.00	2510	0.0015

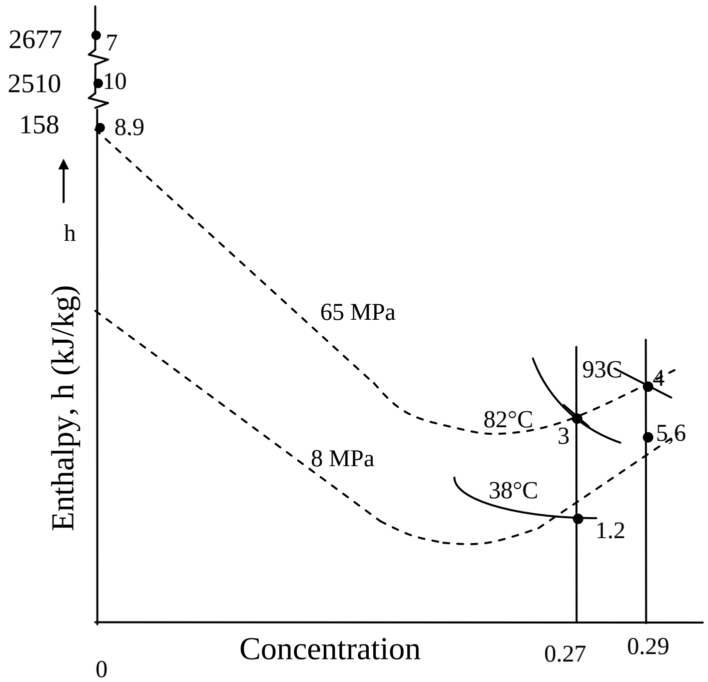

Figure 3.7: Schematic h-x diagram for a typical lithium bromide – water absorption refrigeration cycle

3.3. Thermoeconomic Principles

3.3.1. Thermodynamic Principles

The thermodynamic principles used in the analysis of gas turbine cogeneration systems are the first law of the thermodynamics, entropy balance equation and the exergy balance equation. These equations were applied to the components forming the system. Each of these components were considered as steady state steady flow devices. The kinetic and potential energy and exergy changes in these components were neglected. Under these assumptions these three equations can be written as follows.

First law (conservation of energy):

$$\dot{Q} - \dot{W} = \sum_{e} \dot{m}_e h_e - \sum_{i} \dot{m}_i h_i \qquad \textbf{(3.4)}$$

Entropy balance equation:

$$\dot{S}_{gen} = \sum_{e} \dot{m}_e s_e - \sum_{i} \dot{m}_i s_i - \frac{\dot{Q}}{T_R} \tag{3.5}$$

where subscript R denotes a thermal reservoir.

Availability (exergy balance) equation:

$$\dot{E}_D = \sum_{i} \dot{m}_i e_{fi} - \sum_{e} \dot{m}_e e_{fe} + \left(1 - \frac{T_0}{T_R}\right)\dot{Q} - \dot{W} \tag{3.6}$$

where

$$e_f = (h - h_o) - T_o(s - s_o) \tag{3.7}$$

f is flow

3.3.2. Economic principles

The basic equation in this context is the cost balance equation, which for a steady state steady flow component can be written as:

$$\sum_{i} \dot{C}_i + \dot{Z} = \sum_{e} \dot{C}_e \tag{3.8}$$

where,

$\dot{C}$ is the cost rate of an exergy ($/s)

$\dot{Z}$ is the cost rate of the capital investment for the component ($/s)

Cost rate may be expressed in the following forms:

$$\dot{C} = c\dot{E} \tag{3.9}$$

$$\dot{C} = c(\dot{m}e) \qquad \textbf{(3.10)}$$

where,

c is the cost per unit exergy ($/kJ)

e is the specific exergy (kJ/kg)

$\dot{m}$ is mass flow rate (kg/s)

To transform the capital investment CI ($), to cost rate of capital investment it must multiplied with the capital recovery factor (CRF) and divided by the period of operation of the system per year (s/year).

Thus:

$$\dot{Z} = \frac{CRF.CI}{3600.n_H} \qquad \textbf{(3.11)}$$

where

n_H is the number of hours of operation per year.

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \qquad \textbf{(3.12)}$$

i is the interest rate per annum

n economic life of the investment.

4. SIMULATION MODEL

4.1. Introduction

The cogeneration system considered in this thesis is shown in Figure 4.1. It consists of a gas turbine, heat recovery steam generator, a steam turbine and an absorption refrigeration unit. The steam turbine and the absorption refrigeration unit are coupled to the gas turbine system through the heat recovery steam generator. The thermodynamic analysis of this system is given in section 4.2. The economic analysis of the system is given in section 4.3. Two computer programs have been written to do the analysis of this system and is explained in section 4.4.

Table 4.1 shows the mass flow rates, temperatures and pressures for the different states of the system considered. (The data is obtained from calculation).

Table 4.1: Mass flow rates, temperatures and pressures for the different states of the system shown in Figure 4.1, for a compressor pressure ratio of 10 and 10 MW power production

STATE	$\dot{m}$ (kg/s)	P (kPa)	T (K)
1	30.10	101.3	298.1
2	30.10	1013.0	601.9
3	30.10	962.3	850.0
4	30.64	914.2	1520.0
5	30.64	109.9	1004.9
6	30.64	106.6	764.6
7	30.64	101.3	427.0
8	.54	1200.0	298.1
9	.00	.00	.00
10	.00	.00	.00
11	3.88	4000.0	623.0
12	.00	.00	.00
13	3.88	300.0	406.6
14	.00	.00	.0
15	3.88	300.0	406.6
16	3.88	4000.0	407.6
17	.00	.00	.00

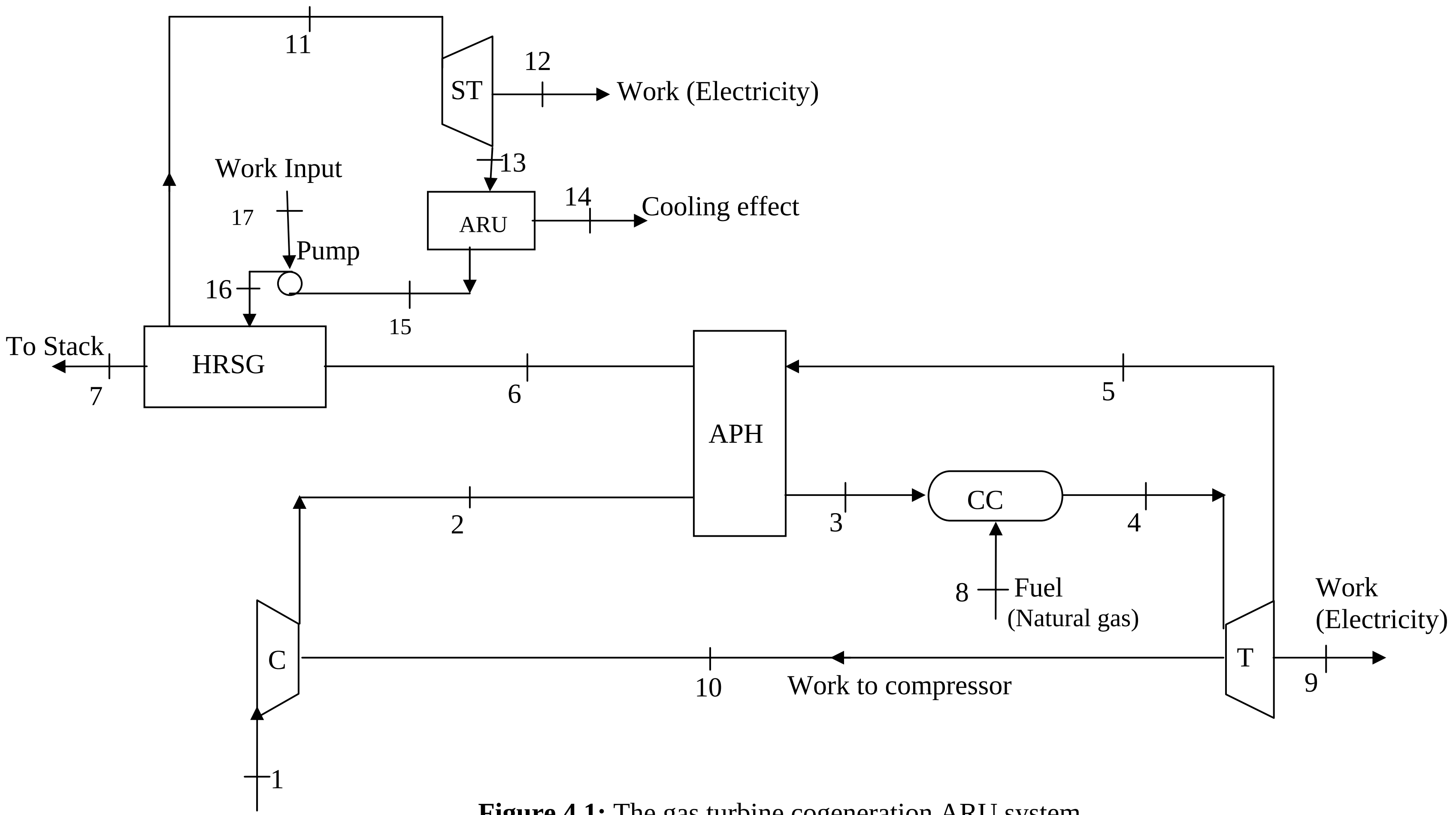

Figure 4.1: The gas turbine cogeneration ARU system

4.2. The Thermodynamic Analysis of the Components

The assumptions underlying the cogeneration system model Figure 4.1 include the following:

a) The cogeneration system operates at steady state.

b) Air and the combustion products are assumed to be ideal gas mixtures.

c) The fuel (natural gas) is taken as methane .

d) Heat transfer from the combustion chamber is 2% of the lower heating value of the fuel.

The thermodynamic analysis of each component of the system is given below as they appear in the flow stream: Compressor, air preheater, combustion chamber, turbine, heat recovery steam generator, steam cycle, absorption refrigeration unit.

4.2.1. Compressor

The air compressor is considered as a steady state steady flow adiabatic device as shown in Figure 4.1. The pressure ratio of the compressor is defined as:

$$r_p = \frac{P_2}{P_1} \tag{4.1}$$

The isentropic efficiency of the compressor is defined as :

$$\eta_c = \frac{T_{2S} - T_1}{T_2 - T_1} \tag{4.2}$$

Where T_{2S} is the temperature at the end of isentropic compression. This temperature is given by :

$$\frac{T_{2S}}{T_1} = r_p^{(k-1)/k} \tag{4.3}$$

where,

$$k = \frac{\bar{c}_p}{\bar{c}_v} \quad \textbf{(4.4)}$$

The specific heat at constant pressure and volume, $\bar{c}_p$ and $\bar{c}_v$ respectively are calculated at the average temperature in the compressor. The $\bar{c}_p$ value in the range between 100 °C and 200 °C can be estimated to within 0.1 % by the following relationship.

$$\bar{c}_p\ (T) = 0.2\ \theta^2 + 1.56\ \theta + 28.48\ (kJ / kmol\text{-}K) \quad \textbf{(4.5)}$$

where,

$$\theta = T / 100. \quad \textbf{(4.6)}$$

Thus, given the inlet state to the compressor and the compressor pressure ratio r_p, the exit temperature T_{2S} was found iteratively by improving $\bar{c}_p$, average and using equation (4.3).

The specific work requirement of the compressor can be found by applying the first law to the compressor :

$$-\bar{w} = \hbar_2 - \hbar_1 \quad \textbf{(4.7)}$$

4.2.2. Air Preheater

Air preheater is considered as a steady state, steady flow device. Effectiveness of the air preheater is defined as :

$$\in_{ap} = \frac{\dot{Q}}{\dot{Q}_{max}} \quad \textbf{(4.8)}$$

where $\dot{Q}_{max}$ is the maximum amount of heat that can be transferred from the exhaust stream to the air and $\dot{Q}$ is the actual amount. Effectiveness, $\in$ of the air preheater has been taken as 95% in this study. Applying the first law to the air preheater yields:

$$\dot{n}_2\left(\hbar_3-\hbar_2\right)= \quad \in_{ap} \dot{n}_5\left(\hbar_5-\hbar_6\right) \tag{4.9}$$

and

$$\hbar_6 = \quad \hbar_5-\frac{\dot{n}_2}{\dot{n}_5 \in_{ap}}\left(\hbar_3-\hbar_2\right) \tag{4.10}$$

T_6 corresponding the $\hbar_6$ is found by trial and error.

4.2.3. Combustion chamber

The combustion process is assumed to occur as a steady state, steady flow process and the fuel is taken as methane (CH_4). The flow diagram of the process is shown in Figure 4.2.

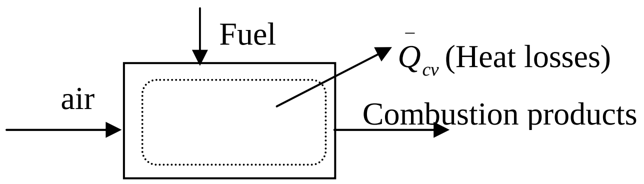

Figure 4.2: Flow diagram of the combustion process

The air supplied to the combustion chamber is assumed to be an ideal gas mixture and has the following molar composition:

Nitrogen (N_2)	0.7784
Oxygen (O_2)	0.2059
Carbon dioxide (CO_2)	0.0003
Water vapour (H_2O)	0.0190

The combustion equation is :

$$
\begin{aligned}
&\lambda CH_4 + \left[0.7748N_2 + 0.2059O_2 + 0.0003CO_2 + 0.019H_2O\right] \\
&\rightarrow (\lambda + 0.0003)\, y_{cc} CO_2 + (\lambda + 0.0003) \quad (1 - y_{cc}) \ CO \\
&+ \left[0.20605 - 0.5\lambda \quad (3 + y_{cc}) - 0.00015 y_{cc}\right] \ O_2 \\
&+ (2\lambda + 0.019)\, H_2O + 0.7748N_2
\end{aligned}
\qquad \textbf{(4.11)}
$$

where; λ is the molar fuel air ratio, y_{cc} is the molar percentage of the carbon in the fuel which is converted to CO_2. y_{cc} is 1 for complete combustion.

λ can be calculated from the first law if the temperature of the combustion products, state of the inlet air, y_{cc} and the heat losses from the combustion chamber, $\overline{Q}_{CV}$ are given. The first law for the combustion chamber can be written as:

$$\overline{Q}_{CV} + \overline{H}_R = \overline{H}_P \qquad \textbf{(4.12)}$$

In the model forming the basis of the computer program, the heat losses were assumed to be 2% of the lower heating value of the fuel and y_{cc} was taken as 1.

The enthalpies and entropies of the substances taking part in the combustion process were calculated by using Table 4.2 and Table 4.3 and equations 4.13 to 4.16 which are given below:

Table 4.2: Specific heat, enthalpy, absolute entropy, and Gibbs function with temperature 298.15 K and 101.325 kPa for various substances in units of kJ/kmol or kJ/kmol – K. Knacke et al. (1991)

1. At Tref = 298.15 K(25°C), Pref = 101.325 kPa					
Substance	Formula	$\overline{c}_p^\circ$ (kJ/kmol-K)	$\overline{h}^\circ$ (kJ/kmol)	$\overline{s}^\circ$ (kJ/kmol-K)	$\overline{g}^\circ$ (kJ/kmol)
Nitrogen	N_2(g)	28.49	0	191.610	-57128
Oxygen	O_2 (g)	28.92	0	205.146	-61164
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	-285879	69.948	-306685
Mehane	CH_4 (g)	35.05	-74872	186.251	-130403

2. For $298.15 < T \le T_{max}$ P_{ref} = 1 bar, with $y = 10^{-3}$ T

$$\bar{c}_p^{o} = a + by + cy^{-2} + dy^{2} \quad \textbf{(4.13)}$$

$$\bar{h}^{\circ} = 10^{3}\left[H^{+} + ay + \frac{b}{2}y^{2} - cy^{-1} + \frac{d}{3}y^{3}\right] \quad \textbf{(4.14)}$$

$$\bar{S}^{\circ} = S^{+} + a\ln T + by - \frac{c}{2}y^{-2} + \frac{d}{2}y^{2} \quad \textbf{(4.15)}$$

$$\bar{g}^{\circ} = \bar{h}^{\circ} - T\bar{s}^{\circ} \quad \textbf{(4.16)}$$

Table 4.3: Constants H^{-}, S^{-}, a, b, c and d required by equations (4.13-16) in Table 4.2. Knacke et al. (1991)

Substance	Formula	H^{+}	S^{+}	a	b	c	d
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
Carbon monoxide	CO(g)	-120.809	18.937	30.962	2.438	-0.280	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

Combustion products are assumed to form an ideal gas mixture. Mole fractions of the constituents of the combustion products are calculated by using equation 4.11.

Enthalpy, entropy and physical exergy of the combustion products are calculated by the following equations:

$$\bar{\hbar}_P = \sum_{i=1}^{N} y_{ni}\hbar_i \quad \textbf{(4.17)}$$

$$\bar{s}_P = \sum_{i=1}^{N} y_{ni}\bar{s}_i \quad \textbf{(4.18)}$$

$$\bar{s}_i = \bar{s}_{io} - \bar{R}_{\ln}\frac{P}{P_0} \quad \textbf{(4.19)}$$

$$\overline{e}_{i,PH} = \left(\hbar_i - \hbar_{i,0}\right) - T_0(\overline{s}_i - \overline{s}_{i,0}) \tag{4.20}$$

$$\dot{e}_P = \sum_{i=1}^{N} y_{ni}\overline{e}_{i,PH} \tag{4.21}$$

where,

y_{ni} : mole fraction of constituent.

P_0, T_0 : environmental pressure and temperature.

A special attention must be paid to the combustion products when brought to environmental conditions (298K, 101.3 kPa) for exergy calculations. If the water content of the combustion products is high, condensation may accur. At environmental conditions the partial pressure of the water vapour cannot exceed 3.17 kPa . If condensation occurs the gas and liquid phases of the combustion products must be considered separately. It should also be noted that when condensation occurs the mole fractions of the constituents of the gas phase changes and this was reflected to the calculations.

4.2.4. Gas turbine

The gas turbine is considered as a steady state steady flow adiabatic device as shown in Figure 4.1. The pressure ratio of the turbine is defined as :

$$r_p = \frac{P_5}{P_4} \tag{4.22}$$

The isentropic efficiency of the turbine is defined as :

$$\eta_{st} = \frac{T_4 - T_5}{T_4 - T_{5S}} \tag{4.23}$$

Where T_{5S} is the temperature at the end of the isentropic expansion. This temperature is given by :

$$\frac{T_{5S}}{T_4} = r_P^{(k-1)/k} \qquad \textbf{(4.24)}$$

where,

$$k = \frac{\overline{c}_p}{\overline{c}_v} \qquad \textbf{(4.25)}$$

The specific heat at constant pressure and volume, $\overline{c}_P$ and $\overline{c}_v$ respectively are calculated at the average temperature in the turbine. The $\overline{c}_P$ value in the range between 100°C and 200°C can be estimated to within 0.1% by the following relationship:

$$\overline{c}_P\,(T) = 0.00355 + T_{AVE} + 30.818 \text{ (kJ/kmol-K)} \qquad \textbf{(4.26)}$$

where,

$$T_{AVE} = T_4 - 100 \qquad \textbf{(4.27)}$$

Thus, given the inlet state to the gas turbine and the gas turbine pressure ratio r_p, the exit temperature T_{5S} was found iteratively by improving $\overline{c}_P$, average and using equation (4.24).

The specific work requirement of the turbine can be found by appling the first law to the turbine :

$$\overline{w} = \hbar_4 - \hbar_5 \qquad \textbf{(4.28)}$$

4.2.5. Heat recovery steam generator

Heat recovery steam generator is considered as a steady state steady flow adiabatic device as shown in Figure 4.3.

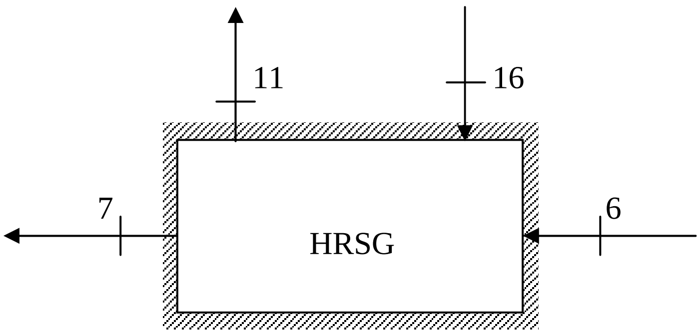

Figure 4.3: Schematic diagram of the heat recovery steam generator

Considering the control volume enclosing the heat recovery steam generator, first law yields:

$$\dot{m}_6(h_6 - h_7) = \dot{m}_{16}(h_{11} - h_{16}) \qquad \textbf{(4.29)}$$

The minimum exit temperature of the combustion products from the heat recovery steam generator is stipulated as 154°C so that condensation of water vapour within the device is prevented. The minimum pinch temperature difference in the heat recovery steam generator is set to 20°C. The mass flow rate of water on the steam cycle side is calculated by considering the first law and the pinch condition. This is illustrated in Figure 4.4 .

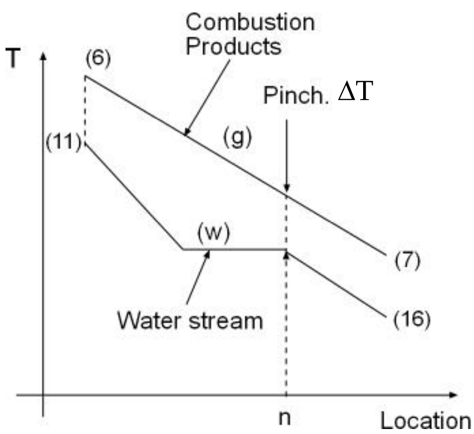

Figure 4.4: Pinch temperature difference in the heat recovery steam generator

The pinch condition restricts the mass flow rate of water so that a minimum temperature difference is kept between the two streams. The first law applied to the heat recovery steam generator before and after the pinch yields:

$$\dot{m}_p C_p \left(T_6 - T_{ng}\right) = \quad \dot{m}_w \left(h_{11} - h_{nw}\right) \tag{4.30}$$

$$\dot{m}_p C_p \left(T_{ng} - T_7\right) = \quad \dot{m}_w \left(h_{nw} - h_{16}\right) \tag{4.31}$$

Mass flow rate of water is determined so that both of the above equations is satisfied and $\Delta T_{pinch} \geq 20$.

4.2.6. Steam turbine cycle

For the steam cycle selected, the turbine inlet conditions are 4 Mpa, 350°C and the turbine exit pressure is 300 kPa. The T-s diagram of the steam cycle is shown in Figure 4.5 and the properties at various states are given in Table 4.4 .

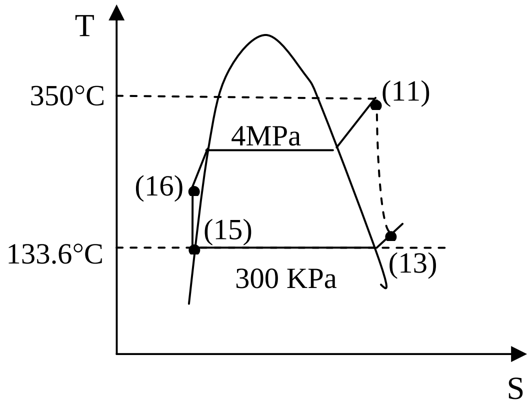

Figure 4.5: Steam cycle part of the system shown on a T-s diagram

Table 4.4: Properties at various states of the steam cycle (Refer to Figure 4.5)

State	T(°C)	P (kPa)	h (kJ(kg)	s (kJ/kg.K)
15	133.6	300	561.5	1.6718
16	134.7	4000	566.1	1.6832
11	350	4000	3092.5	6.5821
13	133.6	300	2638.7	6.7791

The turbine isentropic efficiency was taken as 85%. The net specific work of the cycle was calculated as 449.1 kJ/kg. The heat transfered to the absorption refrigeration system per unit mass of water was calculated as 2077.2 kJ/kg.

4.2.7. Absorbtion refrigeration unit

The absorption refrigeration unit operates with heat given off in the condenser of the steam cycle. The coefficient of performance of the absorption refrigeration unit is given as input to the program. The refrigerating effect of the absorption refrigeration unit is calculated by the following equation:

$$\dot{Q}_R = \dot{m}_w \left(h_{13} - h_{15}\right) \ . \ COP_{ARU} \qquad \textbf{(4.32)}$$

The exergy of the refrigerating effect is defined as the work required to produce the same refrigerating effect and this is:

$$\dot{E}_{14} = \dot{Q}_R \,/\, COP_{VC} \qquad \textbf{(4.33)}$$

Where COP_{VC} is the average COP of the equivalent vapour compression unit.

4.3. Economic Analysis of the Cogeneration Cycle

4.3.1. Cost balance equations

The economic analysis of the cogeneration cycle is done by applying the cost balance equation to each component, specifying the auxiliary equations and giving the external inputs. This yields a set of linear algebraic equations when solved gives the cost rate ($/s) and the cost per unit exergy ($/kJ) of each stream. For a steady state steady flow component of the system the cost balance equation is :

$$\sum_i \dot{C}_i + \dot{Z} = \sum_e \dot{C}_e \qquad \textbf{(4.34)}$$

where

$\dot{C}$ is the cost rate in $/s

$\dot{Z}$ is the cost rate of the capital investment for the component $/s.

The capital investment rate is obtained by multiplying the total capital investment in $ with the capital recovery factor and dividing by the time length of annual operation of the system. Thus,

$$\dot{Z} = \text{CRF.CI} / (3600.n_H) \tag{4.35}$$

As a rule n – 1 auxiliary equations are required if there are n exiting streams. The external inputs in a sense form the boundary conditions of the set of equations. The equations for the components are given below:

Compressor :

$$\dot{C}_1 + \dot{C}_{10} + \dot{Z}_{COMP} = \dot{C}_2 \tag{4.36}$$

$$\dot{C}_1 = 0 \text{ (external input)} \tag{4.37}$$

Air preheater :

$$\dot{C}_2 + \dot{C}_5 + \dot{Z}_{APH} = \dot{C}_3 + \dot{C}_6 \tag{4.38}$$

The auxiliary relation for the air preheater, the purpose of which is to heat the air stream, is that the cost per unit exergy on the hot side remains constant (c_6=c_5). Thus,

$$\frac{\dot{C}_5}{\dot{E}_5} = \frac{\dot{C}_6}{\dot{E}_6} \text{ or } \dot{C}_5 = \frac{\dot{E}_5}{\dot{E}_6} \dot{C}_6 \tag{4.39}$$

Combustion chamber :

$$\dot{C}_3 + \dot{C}_8 + \dot{Z}_{cc} = \dot{C}_4 \tag{4.40}$$

$$\dot{C}_8 = \frac{\dot{m}_8}{\rho_F} C_F \text{ (External input)} \tag{4.41}$$

where

ρ_F is the density of fuel

C_F is the cost per unit volume for the fuel ($/m^3)

Turbine :

$$\dot{C}_4 + \dot{Z}_{TUR} = \dot{C}_5 + \dot{C}_9 + \dot{C}_{10} \tag{4.42}$$

Ignoring the losses during the trasmission of power from the gas turbine to the air compressor, the cost per unit exergy of power is equal i.e ($c_{10} = c_9$). Thus the first auxiliary equation is :

$$\dot{C}_{10} = \frac{\dot{E}_{10}}{\dot{E}_9} \dot{C}_9 \tag{4.43}$$

The other auxiliary relation for the gas turbine is that cost per unit exergy of the stream remains constant ($c_4 = c_5$). Thus the second auxiliary equation becomes:

$$\dot{C}_5 = \frac{\dot{E}_5}{\dot{E}_4} \dot{C}_4 \tag{4.44}$$

Heat – recovery steam generator (HRSG) :

$$\dot{C}_6 + \dot{C}_{16} + \dot{Z}_{HRSG} = \dot{C}_7 + \dot{C}_{11} \tag{4.45}$$

Here the cost per unit exergy of the product stream remains constant ($c_6 = c_7$), thus the auxiliary equation becomes:

$$\dot{C}_7 = 0 \quad \text{or} \quad \dot{C}_7 = \frac{\dot{E}_7}{\dot{E}_6} \dot{C}_6 \tag{4.46}$$

Steam turbine :

$$\dot{C}_{11} + \dot{Z}_{ST} = \dot{C}_{12} + \dot{C}_{13} + \dot{C}_{17} \quad \textbf{(4.47)}$$

The first auxiliary equation specifies that the cost per unit exergy of steam as it flows through the turbine remains constant.

$$\dot{C}_{13} = \frac{\dot{E}_{13}}{\dot{E}_{11}} \dot{C}_{11} \quad \textbf{(4.48)}$$

The second auxiliary equation states that the cost per unit exergy of all work streams are equal.

$$\dot{C}_{17} = \frac{\dot{E}_{17}}{\dot{E}_{12}} \dot{C}_{12} \quad \textbf{(4.49)}$$

Absorption refrigeration unit :

$$\dot{C}_{13} + \dot{Z}_{ARU} = \dot{C}_{14} + \dot{C}_{15} \quad \textbf{(4.50)}$$

The auxiliary equation specifies that the cost per unit exergy of steam passing through the absorption refrigeration unit remains constant.

$$\dot{C}_{15} = \frac{\dot{E}_{15}}{\dot{E}_{13}} \dot{C}_{13} \text{ (auxiliary equation)} \quad \textbf{(4.51)}$$

Pump :

$$\dot{C}_{17} + C_{15} + \dot{Z}_{P} = \dot{C}_{16} \quad \textbf{(4.52)}$$

4.3.2. Capital costs of the components

The overall capital costs of the gas turbine subsystem, steam turbine subsystem and the absorption refrigeration unit were fed to the model as inputs. The break down of the capital costs between the components were as follows:

Gas turbine subsystem :

Compressor	36.6%
Air preheater	10%
Combustion chamber	3.4%
Turbine	35%
Heat recovery steam generator	15%

Steam turbine subsystem :

Turbine	90%
Pump	10%

For the cogeneration system considered in this thesis there are 8 components including the ARU. The external inputs are the cost rates of the fuel and air streams entering the boundaries.

For the calculation of the capital recovery factor, the annual interest rate and the economic life were entered as inputs. Typical values for these variables were 10% and 10 years respectively.

4.4. Implementation of the Numerical Model

Two computer programs were written to realize the computations of the model explained in this chapter. The flow charts of these programs are given in Figure 4.6 and Figure 4.7 . The listing of these programs are given in appendix A and appendix B respectively.

The thermodynamic analysis program does the first law analysis of the system, calculates the mass flow rates of fuel and air, temperatures, pressures and exergy rates at all state points. The input data for this program are the pressure ratio of the compressor, net power of the system, inlet temperature to the combustion chamber and the turbine, pressure drops, efficiencies and investment cost data for all components. This program also prepares the input data for the second program.

The cost analysis program calculates the cost rates and cost per unit exergy at all state points of the system. Input data to this program are the capital cost of components, fuel costs and exergy rates at all state points of the system. This input data is generated in the first program.

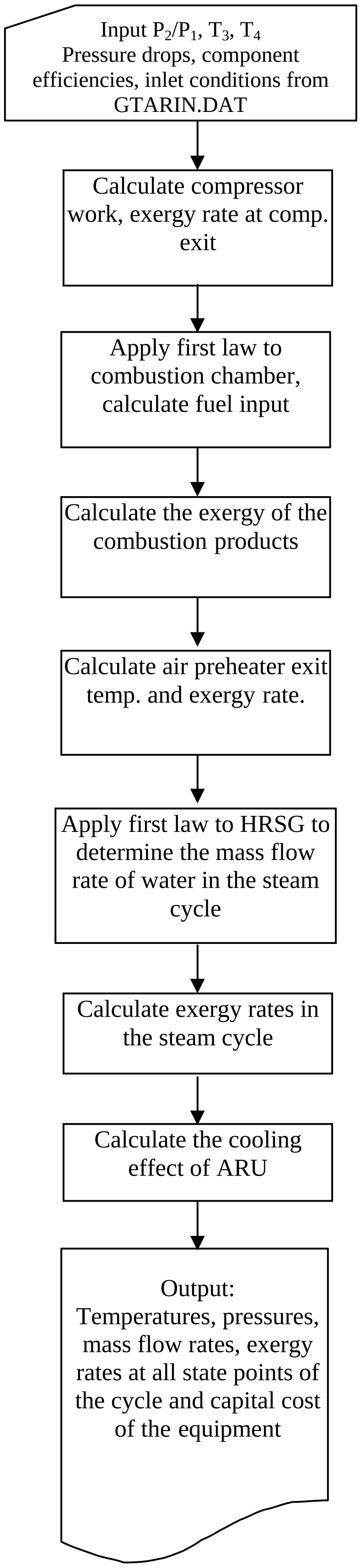

Figure 4.6: Flow chart for the thermodynamic analysis program

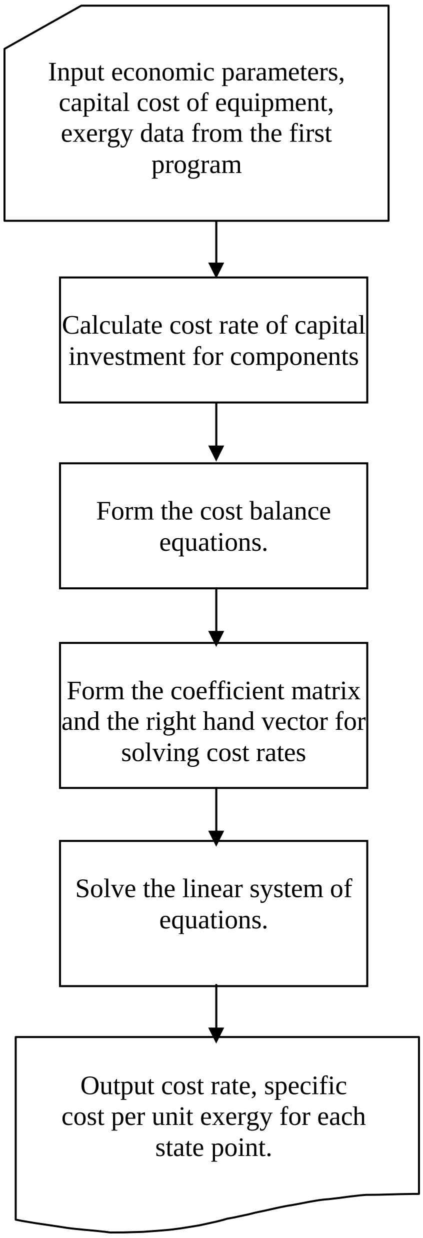

Figure 4.7: Flow chart for the cost analysis program

5. RESULTS AND DISCUSSION

The numerical model was simulated with different values of the decision variables. These are the pressure ratio of the compressor, cost of the natural gas, the investment cost of the gas turbine and the investment cost of the steam turbine. The net power produced in the gas turbine cycle was taken as 10 MW in all simulations.

Range of the decision variables for which simulations were done are given below:

P_2/P_1 : 8 to 12

f_{NG} : 0.15 to 0.25 $/m^3

Z_{GT} : 600 to 700 $/kW

Z_{ST} : 1000 to 1200 $/kW

Exergy rates, cost rates and cost per unit exergy were calculated for all state points (streams) of the system. Exergy destruction, relative cost difference and exergoeconomic factor were calculated for each of the components.

Furthermore an economic analysis was done to determine the pay back period of the system for various values of the decision variables. Results are given and discussed in the tables below.

Table 5.1 shows the mass flow rates, temperatures, pressures and exergy rates for all state points of the system for a compressor pressure ratio of 8. States 1,2 and 3 refer to air. States 4 to 7 represent the combustion products. Flows at 9, 10 and 14 represent the net power, work input to the compressor and the cooling effect respectively. States 11, 13, 15 and 16 are states of the steam cycle.

The decrease of pressure from state 2 to 3 and state 3 to 4 are due to pressure drops in the air preheater and the combustion chamber respectively. On the exhaust side, pressure drops in the air preheater between state 5 and 6 and the heat recovery steam generator between state 6 and 7. The turbine entry and condenser pressures of the steam cycle were chosen as 4 MPa and 300 kPa respectively.

Table 5.1: Exergy rates of the system for a compressor pressure ratio of 8 (Refer to Figure 4.1 for states)

STATE	ṁ kg/s	P (kPa)	T (K)	Exergy Rate (kW)
1	31.43	101.3	298.1	.0
2	31.43	810.4	565.2	8225.4
3	31.43	769.9	850.0	13836.8
4	32.00	731.4	1520.0	34312.4
5	32.00	109.9	1048.7	14536.2
6	32.00	106.6	774.5	7369.6
7	32.00	101.3	427.0	957.1
8	.57	1200.0	298.1	29269.4
9	.00	.0	.0	10000.0
10	.00	.0	.0	8884.0
11	4.23	4000.0	623.0	4805.9
12	.00	.0	.0	1900.6
13	4.23	300.0	406.6	2637.0
14	.00	.0	.0	1883.7
15	4.23	300.0	406.6	287.3
16	4.23	4000.0	407.6	292.6
17	.00	.0	.0	19.8

Table 5.2: Exergy rates of the system for a compressor pressure ratio of 10 (Refer to Figure 4.1 for states)

STATE	ṁ kg/s	P (kPa)	T (K)	Exergy Rate (kW)
1	30.10	101.3	298.1	.0
2	30.10	1013.0	601.9	9051.2
3	30.10	962.3	850.0	13830.1
4	30.64	914.2	1520.0	33454.6
5	30.64	109.9	1004.9	12755.0
6	30.64	106.6	764.6	6835.5
7	30.64	101.3	427.0	916.5
8	.54	1200.0	298.1	28026.0
9	.00	.0	.0	10000.0
10	.00	.0	.0	9721.8
11	3.88	4000.0	623.0	4402.8
12	.00	.0	.0	1741.2
13	3.88	300.0	406.6	2415.8
14	.00	.0	.0	1725.7
15	3.88	300.0	406.6	263.2
16	3.88	4000.0	407.6	268.1
17	.00	.0	.0	18.1

The gas turbine thermal efficiency increased with the pressure ratio. Efficiency is 0.340 for the pressure ratio of 8, 0.357 for the pressure ratio of 10 and 0.365 for the pressure ratio of 12. Since the net power production of the gas turbine cycle is constant at 10 MW, the mass flow rate decreases as the compressor pressure ratio increases. Comparison of Table 5.1, 5.2 and 5.3 yields the following results. Exergy rate is influenced by three variables, namely temperature, pressure and mass flow rate. Changes in these variables are reflected in the exergy rates given in these tables. Specific exergy which can be determined by dividing the exergy rate with the mass flow rate, increases due to increase in pressure as the compressor pressure ratio increases in the gas turbine cycle. However, the decrease in mass flow rate causes the exergy rate to become smaller in Tables 5.2 and 5.3.

Increase of the compressor pressure ratio of the gas turbine cycle decreases both the mass flow rate and the temperature of the exhaust products at state 6. Therefore heat that can be transferred to the steam cycle decreases. For this reason the mass flow rate and the net power output of the steam cycle, as well as the cooling effect of the ARU decrease.

Table 5.3: Exergy rates of the system for a compressor pressure ratio of 12 (Refer to Figure 4.1 for states)

STATE	$\dot{m}$ kg/s	P (kPa)	T (K)	Exergy Rate (kW)
1	29.42	101.3	298.1	.0
2	29.42	1215.6	633.3	9839.2
3	29.42	1154.8	850.0	13982.6
4	29.95	1097.1	1520.0	33179.7
5	29.95	109.9	970.6	11597.7
6	29.95	106.6	759.7	6577.0
7	29.95	101.3	427.0	895.8
8	.53	1200.0	298.1	27394.4
9	.00	.0	.0	10000.0
10	.00	.0	.0	10523.3
11	3.71	4000.0	623.0	4209.9
12	.00	.0	.0	1664.9
13	3.71	300.0	406.6	2310.0
14	.00	.0	.0	1650.1
15	3.71	300.0	406.6	251.6
16	3.71	4000.0	407.6	256.3
17	.00	.0	.0	17.3

5.1. Exergy Destruction in the Components

Exergy destruction is defined by equation (3.6) mentioned earlier. The exergy destructions in the components are due to one or more of the three principal irreversibilities namely combustion, heat transfer and friction.

Table 5.4: Exergy destruction in the components for a compressor pressure ratio of 8 (Refer to Figure 4.1 for states)

Component	Exergy destruction (kW)	Proportion of the total
Compressor	658.6	4.5 %
APH	1555.2	10.7 %
Combustion chamber	8793.8	60.4 %
Gas turbine	892.2	6.1 %
HRSG	1899.2	13.1 %
Steam turbine	268.3	1.8 %
ARU	466	3.2 %
Pump	14.5	0.09 %
Total	14547.8	100 %

The exergy destruction percentage in the components shown in Table 5.4 clearly identify the combustion chamber as the major site of thermodynamic inefficiency. Approximately 60% of the exergy destruction in the cycle occurs here. The percentage of exergy destructions in the heat recovery steam generator and the air preheater are 13.1% and 10.7% respectively. For HRSG and the air preheater heat transfer and friction are the sources of exergy destruction.

Table 5.5: Exergy destruction in the components for a the compressor pressure ratio of 10 (Refer to Figure 4.1 for states)

Component	Exergy destruction (kW)	Proportion of the total
Compressor	670.6	4.9 %
APH	1104.4	8.1 %
Combustion chamber	8401.5	61.6 %
Gas turbine	977.8	7.2 %
HRSG	1784.3	13.1 %
Steam turbine	245.8	1.8 %
ARU	426.9	3.1 %
Pump	13.2	0.09 %
Total	13624.5	100 %

Comparison of Tables 5.4, 5.5 and 5.6 shows that the exergy destruction in the compressor increases as the pressure ratio increases. Exergy destruction in the air preheater decreases as the compressor pressure ratio increases because the mean temperature difference between the air stream and exhaust products decreases in the air preheater. The decrease of exergy destruction in the combustion chamber with increase in the compressor pressure ratio is related solely to change in the mass flow rate, because the inlet and exit temperatures of the combustion chamber is the same for all three cases. In fact the exergy destruction per unit mass flowing through the combustion chamber is nearly constant for all three cases.

Some general observations can be made with respect to exergy destruction. Exergy destruction can be lowered by keeping the temperature differences small during heat transfer processes and minimizing pressure losses in flow processes.

Table 5.6: Exergy destruction in the components for a compressor pressure ratio of 12 (Refer to Figure 4.1 for states)

Component	Exergy destruction (kW)	Proportion of the total
Compressor	684	5.2 %
APH	877.3	6.6 %
Combustion chamber	8197.3	62.1 %
Gas turbine	1058.7	8.0 %
HRSG	1727.6	13.1 %
Steam turbine	235	1.8 %
ARU	408.3	3.1 %
Pump	12.6	0.095 %
Total	13200.8	100 %

5.2. Analysis of the Cost Rates and Cost per Unit Exergy for Each State Point

The cost rates and cost per unit exergy for the state points of the simulation model is given in Tables 5.7 to 5.9. The cost of fuel which is natural gas is 0.2 $\$/m^3$ in Tables 5.7 to 5.9.

Table 5.7: Cost rates and cost per unit exergy of the system for a compressor pressure ratio of 10

State	$\dot{E}$ (kW)	$\dot{C}$ ($/s)	c ($/kJ)
1	.0	.0000	.0000E+00
2	9051.2	.1248	.1378E-04
3	13830.1	.1895	.1370E-04
4	33454.6	.3453	.1032E-04
5	12755.0	.1316	.1032E-04
6	6835.5	.0705	.1032E-04
7	916.5	.0095	.1032E-04
8	28026.0	.1546	.5515E-05
9	10000.0	.1147	.1147E-04
10	9721.8	.1115	.1147E-04
11	4402.8	.0721	.1638E-04
12	1741.2	.0402	.2310E-04
13	2415.8	.0396	.1638E-04
14	1725.7	.0552	.3201E-04
15	263.2	.0043	.1638E-04
16	268.1	.0056	.2099E-04
17	18.1	.0004	.2310E-04

It is observed that the highest unit exergy cost in the gas turbine cycle is achieved at state (2) at the exit of the air compressor. This is because the investment cost of the compressor is high and the driving input is mechanical power. Considering the whole system it was noticed that the cost per unit exergy is considerably higher for state (11) than the net power state (9). This is due to the addition of the heat recovery steam generator which represents an increase in the investment cost. The cost per unit exergy at state (9) is 0.041 $/kWh. The cost per unit exergy at state (12) is 0.083 $/kWh which is greater than the cost per unit exergy at state (9). This is due to additional capital investment in the heat recovery steam generator and the steam turbine. The factors affecting the cost per unit exergy are the investment cost, exergy destruction and fuel cost.

Since work is produced by the gas turbine and the steam turbine and cost per unit exergy of each is different, an average cost of electricity can be found by weighting the costs with the exergy rates. If this is done the cost of electricity produced by the model is found as 0.048 $/kWh. As the fuel cost varied from 0.15 $/m^3 to 0.25 $/m^3 the average cost of electricity varied from 0.038 $/kWh to 0.057 $/kWh. The average industrial cost of electricity in Europe is 0.095 $/kWh.

Table 5.8: Cost rates and cost per unit exergy of the system for a compressor pressure ratio of 8

State	$\dot{E}$ (kW)	$\dot{C}$ ($/s)	c ($/kJ)
1	.0	.0000	.0000E+00
2	8225.4	.1159	.1409E-04
3	13836.8	.1940	.1402E-04
4	34312.4	.3566	.1039E-04
5	14536.2	.1511	.1039E-04
6	7369.6	.0766	.1039E-04
7	957.1	.0099	.1039E-04
8	29269.4	.1614	.5515E-05
9	10000.0	.1155	.1155E-04
10	8884.0	.1026	.1155E-04
11	4805.9	.0782	.1627E-04
12	1900.6	.0437	.2297E-04
13	2637.0	.0429	.1627E-04
14	1883.7	.0600	.3185E-04
15	287.3	.0047	.1627E-04
16	292.6	.0061	.2088E-04
17	19.8	.0005	.2297E-04

Table 5.9: Cost rates and cost per unit exergy of the system for a compressor pressure ratio of 12

State	$\dot{E}$ (kW)	$\dot{C}$ ($/s)	c ($/kJ)
1	.0	.0000	.0000E+00
2	9839.2	.1334	.1356E-04
3	13982.6	.1887	.1349E-04
4	33179.7	.3410	.1028E-04
5	11597.7	.1192	.1028E-04
6	6577.0	.0676	.1028E-04
7	895.8	.0092	.1028E-04
8	27394.4	.1511	.5517E-04
9	10000.0	.1142	.1142E-04
10	10523.3	.1202	.1142E-04
11	4209.9	.0692	.1644E-04
12	1664.9	.0385	.2315E-04
13	2310.0	.0380	.1644E-04
14	1650.1	.0529	.3208E-04
15	251.6	.0041	.1644E-04
16	256.3	.0054	.2104E-04
17	17.3	.0004	.2315E-04

The cost per unit exergy of the cooling effect was found as 0.115 $/kWh for a natural gas cost of 0.2 $/m^3 and pressure ratio of 10 in the gas turbine cycle. If the cooling cost is expressed in terms of per unit energy rather than exergy a value of 0.022 $/kWh is found. The calculation is shown in Appendix C.

Variation of the pressure ratio of the gas turbine cycle has little effect on the cost of the cooling effect. A 50% change in the fuel cost, causes a change of 25% in the cost of cooling.

5.3. Determination of the Relative Cost Difference of the Components

The relative cost difference is defined as the relative increase in the average cost per unit exergy between fuel and product in a component, Bejan et. al (1996).

$$r_k = \frac{c_{Pk} - c_{Fk}}{c_{Fk}} \tag{5.1}$$

where,

c_{Pk} = cost per unit exergy of the product

c_{Fk} = cost per unit exergy of the fuel

Cost increase of a stream as it passes through a component is caused by two factors. First one is the exergy destruction which is related to the thermodynamic performance of a component. Second factor is the investment and maintenance costs. Table 5.10 gives the relative cost difference values for the components of the simulation model. It is seen that the higher relative cost differences are associated with the ARU, steam turbine , the HRSG and the APH in that order. This implies that improvements in thermodynamic performance and reduction in investment costs should first be achieved in these components. It was observed that change in the pressure ratio of the gas turbine cycle or the cost of the fuel did not change this order. In general, improvements in thermodynamic performance is accompanied by an increase in capital investment. Therefore it is important to know the relative importance of these factors in increasing the cost.

The exergoeconomic factor (f_k) is a parameter that gives an indication of this.

Table 5.40: The relative cost difference for the components (r_k). Cost of fuel is considered to be 0.2 \$/m^3.

Relative cost difference, r_k		
	Pressure Ratio	
Component	8	10
Compressor	0.216	0.19
APH	0.33	0.31
Combustion chamber	0.259	0.254
Gas turbine	0.110	0.110
HRSG	0.346	0.360
Steam turbine	0.415	0.403
ARU	0.96	0.96
Pump	0.256	0.22

5.4. Determination of the Exergoeconomic Factor of the Components

The exergoeconomic factor (f_k) is defined as the ratio of investment cost to total cost, Bejan et. al (1996)

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{Fk}.\dot{E}_{Dk}} \qquad \textbf{(5.2)}$$

where,

$\dot{E}_{Dk}$ is the rate of exergy destruction in kW.

c_{Fk} is the cost per unit exergy of the fuel in \$/kJ.

$\dot{Z}_k$ is the rate of capital investment including the operation and maintenance costs.

Table 5.11 gives the exergoeconomic factors of the components of the simulation model. It is observed that the exergoeconomic factor is high for the ARU, the compressor, the pump, the steam turbine and the gas turbine. This indicates that investment and maintenance costs are more influential in the relative cost increase in these components. Noting that the ARU was the component with the highest exergoeconomic factor also, it can be concluded that lowering the investment cost in that component at the expense of thermodynamic performance may result in overall cost reduction. Similarly low exergoeconomic factors for the APH and the HRSG suggest that improving the thermodynamic performance even though this will require higher investment, may reduce the overall cost of the model.

Table 5.51: The exergoeconomic factor of the components (f_k). Cost of fuel is considered to be 0.2 $/m^3

Exergoeconomic factor, (f_k)		
	Pressure Ratio	
Component	8	10
Compressor	0.626	0.628
APH	0.325	0.364
Combustion chamber	0.014	0.014
Gas turbine	0.55	0.546
HRSG	0.158	0.174
Steam turbine	0.666	0.615
ARU	0.739	0.72
Pump	0.88	0.756

5.5. Calculation of the payback period

The payback period, pp is defined as the length of time required for the cash inflows received from a project to recover the initial investment.

$$pp = \frac{\text{Total depreciable investment}}{\text{Annual net profit}} \tag{5.3}$$

The total depreciable investment is the difference between the initial capital investment and the salvage value.

An example of the calculation of the pay back period is given in appendix C. Table 5.12 shows the pay back period for the different parameters. The examination of Table 5.12 shows that the increase in fuel cost increases the annual expenditure and hence decreases the annual net profit, resulting in an increase in the pay back period. The increase in the investment cost results in an increase in the pay back period. The increase in the pressure ratio decreases the pay back period as the system becomes more efficient, less fuel is consumed and the annual net profit is increased. The payback period for the adopted system was found to be between 7 and 9 years for different parameters shown in Table 5.12. While the average value given for Europe in the literature is 12 years, Colonna and Gabrielli (2003).

Table 5.62: Pay back period for 10 MW power production for different parameters

Case	$\frac{P_2}{P_1}$	f_{NG} $/m^3	CI_{GT} $	CI_{ARU} $	CI_{ST} $	Pay back period (years)
1	10	0.15	7000000	3870000	1740000	6.93
2	10	0.20	7000000	3870000	1740000	8.99
3	10	0.25	7000000	3870000	1740000	8.05
4	8	0.20	7000000	4220000	1900000	8.06
5	10	0.20	6000000	3870000	1740000	8.28
6	12	0.20	7000000	3870000	1660000	8.89
7	10	0.20	7000000	3870000	2090000	7.95

5.6. Conclusions and Recommendations

A model for a cogeneration system which produces electricity and cooling effect (refrigeration) was proposed in this thesis. The model consists of a combined cycle (gas and vapour power cycles) driving an absorbtion refrigeration unit. The thermodynamic analysis of the model was made for different pressure ratios of the gas turbine cycle. The economic analysis of the model was made for different investment costs for the components and fuel costs. It was shown that the cost of electricity that can be produced by such a system, would vary between 0.04 and 0.06 $/kWh, and the cost of the cooling effect would vary between 0.018 and 0.026 $/kWh. These values compare favorably with the current costs of these commodities in the market.

The principal advantage of cogeneration is to enable more effective use of fuel. The fuel utilization effectiveness of the proposed system is 70%. If the same amount of electricity and cooling effect were to be produced separately approximately 40% more fuel would have to be utilized. Therefore the use of these cogeneration systems in sectors such as food processing and tourism will produce economic benefits for countries with hot climates.

Finally some follow up studies to this thesis may be recommended. The exergoeconomic analysis used in this thesis may be used for thermal system optimization. For this study a detailed thermodynamic performance and cost data base for components forming the system will be needed. The system proposed is flexible in the sense that more electricity may be produced at the expense of the cooling effect and vice versa. Therefore the transient operation of these systems under different electricity and refrigeration demands may be studied. Absorption refrigeration systems which have higher COP and are more adoptable to cogeneration need to be further studied. These may be multistage absorption refrigeration systems or systems using different binary mixtures.

REFERENCES

Adewusi, S. and Zubair, M., 2004. second Law Based Thermodynamic Analysis of Ammonia-Water Absorption Systems. *Energy. Convers. Mgmt. Vol. 45, pp.2355-2369.*

Aphornratana, S., 1995. Theoretical and Experimental Investigation of a Combined Ejector-Absorption Refrigerator, Ph.D. thesis, University of Sheffield, UK.

Apte, A., 1999. Efficient and Economical Storages Driven by ARP, website.

Bassols, J., Kuekellkom, B., Langreck, J., Schneider, R. and Veelken, H., 2002. Trigeneration in the Food Industry. *Applied Thermal Engineering, Vol.22, pp 595-602.*

Bejan, A., Tasatsaronis, G. and Moran, M., 1996. Thermal Design and Optimization. John. Wiley and Sons, Inc, New York.

Bilgen, E., 2000. Exergetic and Engineering Analyses of Gas Turbine Based Cogeneration System. *Energy, Vol. 25, pp 1215-1229*

Colonna, P. and Gabrielli, S., 2003. Industrial Trigeneration Using Ammonia-Water Absorption Refrigeration Systems (AAR). *Applied Thermal Engineering, Vol. 23, pp 381.*

Cengel, Y. and Boles, M., 1998. Thermodynamic an Engineering Approach. McGRAW-HILL Book Company. New York

Derbentli, T., 2002. Thermal Environmental Engineering Lecture Notes, Istanbul Technical University.

Devault, R. and Mrsalla, J., 1990. Ammonia-Water Triple Effect Absorption Cycle. *ASHRAE Trans, Vol. 96, pp 672-82.*

El-sayed, Y., 1992. Heat Pump Assisted Cogeneration in Process Industries. *AES. Vol. 27/HTD- Vol.228.Thermodynamic and the Design Analyis, and Improvement of Energy Systems ASME.*

Gosney, W., 1982. Principle of Refrigeration Cambridge Uni. Press.

Grossman, G., Zaltash, A., Adock, P. and Devault, R., 1995. Simulating a-4 Effect Absorption Chiller, *ASHRAE J., Jun., pp 45-53.*

Guarinello, F., Cerqueira, S. and Nebra, S., 2000. Thermodynamic Evaluation of a Gas Turbine Cogeneration System. *Energy Convers. Mgmt. Vol, 41, pp 1191-1200.*

Herold, K. and Redermacher, L., 1998. Absorption Heat Pump, *Mech. Eng, Aug, pp 68-73.*

Huang, F., 1990. Performance Evaluation of Selected Combustion Gas Turbine. Cogeneration System Based on First and Second-Law Analysis. Department of Mechanical Engineering. San. Jose State University. San Jose. CA95192.

Kartchenko, N.V., Taylor, H. and Francis, K., 1998**.** Advanced Energy Systems. Washington, Chapter Seven, pp 163-182.

Knacke, O., Kubashchevski, O. and Hesselmann, K., 1991, Thermochemical Properties of Inorganic Substances Second Ed. Springer – Verlag, Berlin.

Kwak, H., Kim, D. and Jean, J., 2003. Exergetic and Thermodynamic Analyses of Power Plants. *Energy, Vol. 28, pp 343-360.*

Laune, H., 2002. Regional Report Europe: Heat Pumps - Status and Trends. *Int J Refrig; Vol. 25, pp 414-420.*

Mısra, R., Sahor, P. and Gupta, A., 2002. Application of the Exergetic Cost Theory to the LiBr / H_2O Vapour Absorption System. *Energy, Vol. 27, pp1009-1025.*

Mısra, R., Sahor, P., Sahoo, S. and Gupta, A., 2003. Thermoeconomic Optimization of a Single Effect Water/LiBr Vapour Absorption Refrigeration System. *Int J Refrg; Vol. 26, pp 158-169.*

Mone, C., Chau, D. and Phelan, P., 2001. Economic Feasibility of Combined Heat and Power and Absorption Refrigeration with Commercially Available Gas Turbines. *Energy Convers Mgmt. Vol. 42, pp 1559-1573.*

Moran, M. and Shapiro, H., 1996. Fundamentals of Engineering Thermodynamics. John. Wiley and Sons, Inc. New York.

Rice, G., 1987. Thermodynamic Evaluation of Gas Turbine Cogeneration Cycles: Part 1-Heat Balance Method Analysis. *Journal of Engineering for Gas Turbines and Power. ASME. Spiring, TX 77383 Fellow.*

Sabri, D. and Pierpaoli, P., 2002. Optimization of Inlet Air-Cooling Systems for Steam Injected Gas Turbines. *International Journal of Thermal Sciences. Vol. 41, pp 815-822.*

Saghiruddin, S. and Altamush, M., 2001. Economic Analysis of Two Stage Dual Fluid Absorption Cycle for Optimizing Generator Temperatures. *Energy Convers Mgmt. Vol. 42, pp 407-437.*

Schinazaki, Y., 2003. Evaluation of Refrigerating and Air-Conditioning Technologies in Heat Cascading System under the Carbon Dioxide Emissions Constraint: The Proposal of The Energy Cascade Balance Table. *Energy Policy, Vol. 31, pp 1685-1697*

Siddiqui, M., 1997. Economic Analysis of Absorption Systems: Part A- Design And Cost Evaluation. *Energy Convers. Mgmt Vol. 38 No.93 pp889-904.*

Siddiqi, M., 1997. Economic Analysis of Absorption System: Part b- Otimization of Operating Parameters. *Energy Convers. Mgmt. Vol. 38, No. 9, pp 905-918*

Silveria, L., Beyene, A., Leal, E., Santana, J. and Okada, D., 2003. Thermoeconomic Analysis of a Cogeneration System of a University Campus. *Applied Thermal Engineering, Vol. 22, pp 1471-1483.*

Srikhirin,P., Aphornratana, S. and Chunpaibulpatana, S., 2001. A Review of Absorption Refrigeration Technologies. *Renewable and Sustainable Energy Reviews, Vol. 5, pp 343-372.*

Sun, D., 1997. Thermodynamic Design Data and Optimum Design Maps for Absorption Refrigeration Systems. *Applied Thermal Engineering Vol. 17, No.3, pp 211-221.*

Threlkeld, L., 1970. Thermal Environmental Engineering. Prentic-Hall,INC. Englewood Cliffs, New Jersey.

Usta, N. and Ileri, A., 1999. Computerised Economic Optimization of Refrigeration System Design. Energy Convers. Mgm., *Vol. 40, pp 1089-1109.*

White, D. and Oneil, B., 1995. Analysis of an Improved Aqua-Ammonia Absorption Refrigeration Cycle Employing Evaporator Blowdown to Provide Rectifier Reflux. *Applied Energy, Vol. 50, pp 523-337.*

Ziegler, B. and Trepp, C., 1984. Equation of State for Ammonia-Water Mixtures. *Volume 7. Number 2. Butterworth and Company (Publishers)*

Ziegler, F., Kahn, R., Summerer, F. and Alefeld, G., 1993. Multi- Absorption Chillers. *Int J Refrg; Vol. 16 (5), pp 301-310.*

APPENDIX A

The thermodynamic analysis program

The thermodynamic analysis program does the first law analysis of the system, calculates the mass flow rates of fuel and air, temperatures, pressures and exergy rates at all points of the system. It is written in Fortran. Since the listing of this program is longer than 10 pages, it has been given on the attached Fortran File in this CD.

Input data for the thermodynamic analysis is given below:

```
ANALYSIS OF THE GAS TURBINE ARU TRIGENERATION SYSTEM
101.3     298.15    10.0      10000.0
 850.0    1520.0
0.05      0.05      0.03      0.05
1200.0    298.15
0.86      0.86      0.95      1.00
1.000     427.0     0.60
0.100     10.0      700.0     800.0     1000.0    0.2
```

First row is the title.

Second row is the inlet pressure of the compressor in kPa, the inlet temperature of the compressor in K, the pressure ratio P_2/P_1 and the power output of the turbine in kW respectively.

Third row is the combustion chamber inlet temperature in K, the turbine inlet temperature in K respectively.

Fourth row is the pressure drop of the air preheater on the air side, the pressure drop of the combustion chamber, the pressure drop of the air preheater on the gas side and pressure drop of the heat recovery steam generator respectively.

Fifth row is the inlet pressure of the fuel (methane) in kPa and the inlet temperature of the fuel in K.

Sixth row is the efficiency of the compressor, the efficiency of the turbine, effectiveness of the air preheater and effectiveness of the heat recovery steam generator respectively.

Seventh row is the quality x of the steam, the exhaust temperature in K and the coefficient of performance of the ARU respectively.

Eighth row is the interest rate, economic life of the investment, the specific cost of the gas turbine in $/kW, the specific cost of the ARU in $/kW, the specific cost of the steam turbine in $/kW and cost of fuel in $/m^3 respectively.

Output data for the thermodynamic analysis program for a compressor ratio of 8 is given below:

STATE	MDOT(kg/s)	P(kPa)	T(K)	EX RATE(kW)
1	31.43	101.3	298.1	.0
2	31.43	810.4	565.2	8225.4
3	31.43	769.9	850.0	13836.8
4	32.00	731.4	1520.0	34213.4
5	32.00	109.9	1048.7	14536.2
6	32.00	106.6	774.5	7369.6
7	32.00	101.3	427.0	957.1
8	.57	1200.0	298.1	29269.4
9	.00	.0	.0	10000.0
10	.00	.0	.0	8884.0
11	4.23	4000.0	623.0	4805.9
12	.00	.0	.0	1900.6
13	4.23	300.0	406.6	2637.0
14	.00	.0	.0	1883.7
15	4.23	300.0	406.6	287.3
16	4.23	4000.0	407.6	292.6
17	.00	.0	.0	19.8

First column is the number of state points of the system.

Second column is the mass flow rates at different state points of the system in kg/s.

Third column is the pressures at different state points of the system in kPa.

Fourth column is the temperatures at different state points of the system in degree K.

Fifth column is the exergy rates at different state points of the system in kW.

APPENDIX B

The cost analysis program

```
C      PROGRAM : ABMT07.FOR
C      THIS PROGRAM WAS WRITTEN TO FORM THE
C      COEFFICIENT MATRIX AND THE RIGHT HAND VECTOR
C      TO SOLVE THE SET OF EQUATIONS FOR THE COST RATES
C      OF THE STREAMS IN THE GAS TURBINE ARU SYSTEM.
C      CALCULATION OF COST OF EQUIPMENT ( ZC VALUES)
C      WAS INCORPORATED TO ABMT07.
C      c A.M.El Hannan and T. Derbentli, July 24, 2005
C                 Revised, August 21, 2005
C                 Revised, August 27, 2005
C      ************************************************
       CHARACTER*6 BASLIK
       DIMENSION BASLIK(10)
       DIMENSION A(17,17),B(17),X(17),E(17),SC(17),ZC(8)
C      ***************************************************
C      COST PERCENTAGES OF THE GT AND ST CYCLE COMPONENTS
C      ARE GIVEN IN THE FOLLOWING DATA STATEMENTS
C      ***************************************************
       DATA CPCOMP,CPAPH,CPCC,CPTUR,CPHRSG/0.366,0.1,0.034,
      -0.35,0.15/
       DATA CPSTUR,CPPUMP/0.9,0.1/
       OPEN(8,FILE='AMEXER.DAT',STATUS='OLD')
       OPEN(6,FILE='COROUT.DAT',STATUS='NEW')
C      ***************************************************
C      DENSITY (ROHF) OF NATURAL GAS IS TAKEN AS 0.7 kg/m3
C      N IS THE NUMBER OF FLOW STREAMS,
C      NCOMP IS THE NUMBER OF COMPONENTS IN THE SYSTEM.
C      ***************************************************
       ROHF=0.7
       N=17
       NCOMP=8
       DO 2 I=1,N
       X(I)=0.01
    2  CONTINUE
C      ******************************************
C      INPUT DATA IS READ FROM THE FILE AMEXER.DAT
C      RINT : INTEREST RATE, EN : ECONOMIC LIFE
C      COSTGT,COSTARU,COSTST : CAPITAL COSTS OF
C      GAS TURBINE, ABSORBTION REF UNIT AND STEAM
```

```
C       TURBINE RESPECTIVELY IN $.
C       EMDOTF : MASS FLOW RATE OF FUEL (NG)
C       ZF : COST OF FUEL IN $/m3.
C       E(I) : EXERGY AT STATE I CALCULATED IN THE
C       PREVIOUS PROGRAM AND WRITTEN TO AMEXER.DAT
C       I VARIES FROM 1 TO N WHICH IS 17.
C       ******************************************
        READ(8,210)BASLIK
  210   FORMAT(10A6)
        WRITE(6,300)BASLIK
  300   FORMAT(8X,10A6,/8X,'c (2005) A. Elhannan, T. Derbentli',/)
        READ(8,212)RINT,EN
  212   FORMAT(F10.3,F5.1)
        WRITE(6,310)RINT,EN
  310   FORMAT(8X,'INTEREST RATE IS',F5.2,' pa,  ECONOMIC LIFE',
       -' IS',F5.1,' YEARS')
        READ(8,214)COSTGT,COSTARU,COSTST
  214   FORMAT(3E10.3)
        WRITE(6,320)COSTGT,COSTARU,COSTST
  320   FORMAT(8X,'CAPITAL COSTS OF THE GT, ARU AND',
       -' THE ST ARE : ',/9X,3F12.1,'  $')
        READ(8,216)EMDOTF,ZFUEL
  216   FORMAT(2F10.3)
        WRITE(6,324)EMDOTF,ZFUEL
  324   FORMAT(8X,'MASS FLOW RATE OF FUEL : ',F6.4,' kg/s',
       -/,8X,'COST OF FUEL           : ',F4.2,' $/m3',/)
        READ(8,220)(E(I),I=1,N)
  220   FORMAT(6F10.1)
C       ******************************************
        CONMIL=1.0E-06
        COSTGT=COSTGT*CONMIL
        COSTARU=COSTARU*CONMIL
        COSTST=COSTST*CONMIL
        TER1=(1.0+RINT)**EN
        CRF=RINT*TER1/(TER1-1.0)
        FACTOR=CRF/(8.76*3.6)
C       **********************************************
C       CALCULATION OF THE ZC (COST RATE) VALUES
C       FOR THE COMPONENTS OF THE TRIGENERATION SYSTEM.
C       **********************************************
        TERM1=COSTGT*FACTOR
        ZC(1)=TERM1*CPCOMP
        ZC(2)=TERM1*CPAPH
        ZC(3)=TERM1*CPCC
        ZC(4)=TERM1*CPTUR
        ZC(5)=TERM1*CPHRSG
        TERM2=COSTST*FACTOR
        ZC(6)=TERM2*CPSTUR
        ZC(7)=COSTARU*FACTOR
        ZC(8)=TERM2*CPPUMP
```

```
C     ******************************************************
C     CALCULATION OF THE ELEMENTS OF THE COEFFICIENT
C     MATRIX (A) AND THE RIGHT HAND VECTOR (b) IN A x = b ,
C     x IS THE VECTOR REPRESENTING THE COST RATES IN $/s
C     ******************************************************
      DO 4 I=1,N
      DO 6 J=1,N
      A(I,J)=0.0
    6 CONTINUE
      B(I)=0.0
      A(I,I)=1.0
    4 CONTINUE
      A(2,1)=-1.0
      A(2,10)=-1.0
      B(2)=ZC(1)
      A(3,2)=-1.0
      A(3,5)=-1.0
      A(3,6)=1.0
      B(3)=ZC(2)
      A(4,3)=-1.0
      A(4,8)=-1.0
      B(4)=ZC(3)
      A(5,4)=-E(5)/E(4)
      A(6,5)=-E(6)/E(5)
      A(7,6)=-E(7)/E(6)
      B(8)=EMDOTF*ZFUEL/ROHF
      A(9,4)=-1.0
      A(9,5)=1.0
      A(9,10)=1.0
      B(9)=ZC(4)
      A(10,9)=-E(10)/E(9)
      A(11,7)=1.0
      A(11,6)=-1.0
      A(11,16)=-1.0
      B(11)=ZC(5)
      A(12,11)=-1.0
      A(12,13)=1.0
      A(12,17)=1.0
      B(12)=ZC(6)
      A(13,11)=-E(13)/E(11)
      A(14,13)=-1.0
      A(14,15)=1.0
      B(14)=ZC(7)
      A(15,13)=-E(15)/E(13)
      A(16,15)=-1.0
      A(16,17)=-1.0
      B(16)=ZC(8)
      A(17,12)=-E(17)/E(12)
C     *******************************************
      W=1.0
```

```
      ERTOP=0.000006*N
      NITER=0
C     ITERASYON BASLIYOR
C     *******************
   52 HATOP=0.0
      NITER=NITER+1
      DO 70 I=1,N
      ABSA=ABS(A(I,I))
      IF(ABSA.LT.0.00001)WRITE(*,*)'A(I,I) SIFIR'
      TOPA=0.
      DO 72 J=1,N
      TOPA=TOPA+A(I,J)*X(J)
   72 CONTINUE
   80 XOLD=X(I)
      X(I)=X(I)+W*(B(I)-TOPA)/A(I,I)
      HATOP=HATOP+ABS(X(I)-XOLD)
   70 CONTINUE
      IF(HATOP.LT.ERTOP)GO TO 100
      IF(NITER.GT.50)GO TO 90
      GO TO 52
   90 WRITE(*,*)'CONVERGENCE IS NOT ACHIEVED'
  100 WRITE(6,330)
  330 FORMAT(8X,' #      E (kW)   ','    C ($/S) ','   c($/kJ)  ',/)
      DO 104 I=1,N
      IF(E(I).LT.0.00001)GO TO 106
      SC(I)=X(I)/E(I)
      GO TO 108
  106 SC(I)=0.0
  108 WRITE(6,340)I,E(I),X(I),SC(I)
  340 FORMAT(8X,I2,F12.1,F12.4,E14.4)
  104 CONTINUE
      STOP
      END
```

Input data for the cost analysis program is given below:

```
        ANALYSIS OF THE GAS TURBINE ARU TRIGENERATION SYSTEM
.100        10.0
.7E+07      .422E+07    .190E+07
0.57        0.20
.0          8225.4      13836.8     34312.4     14536b2     7369.6
957.1       29269.4     10000.0     8884.0      4805.0      1900.6
2637.0      1883.7      287.3       292.6       19.8
```

First row is the title.

Second row is the interest rate and the economic life respectively

Third row is the cost of the gas turbine cycle in dollars, the cost of ARU in dollars and the cost of steam turbine cycle in dollars respectively.

Fourth row is the mass flow rate of fuel in kg/s and the cost of fuel in $\$/m^3$ respectively.

Fifth row is the exergy rates of different states from 1 to 17 as shown by Figure 4.1 respectively.

Output data fort he cost analysis program for a compressor ratio of 8 is given below:

STATE	E(kW)	C($/s)	c($/kJ)
1	.0	.0000	.0000 E+04
2	8225.4	.1159	.1409 E-04
3	13836.8	.1940	.1402 E-04
4	34312.4	.3556	.1039 E-04
5	14536.2	.1511	. 1039 E-04
6	7369.6	.0766	. 1039 E-04
7	957.1	.0099	. 1039 E-04
8	29269.4	.1614	.5515 E-05
9	10000.0	.1155	.1155 E-04
10	8884.0	.1026	.1155 E-04
11	4805.9	.0782	.1627 E-04
12	1900.6	.0437	.2297 E-04
13	2637.0	.0429	.1627 E-04
14	1883.7	.0600	.3185 E-04
15	287.3	.0047	.1627 E-04
16	292.6	.0061	.2088 E-04
17	19.8	.0005	.2297 E-04

First column is the number of state points of the system.

Second column is the exergy rates at different state points of the system in kW.

Third column is the cost rates at different state points of the system in $/s.

Fourth column is the cost per unit exergy at different state points of the system in $/kJ.

APPENDIX C

1. Determination of the cost of the cooling effect per unit energy.

Case (a)

This case considers the system adopted in this thesis.

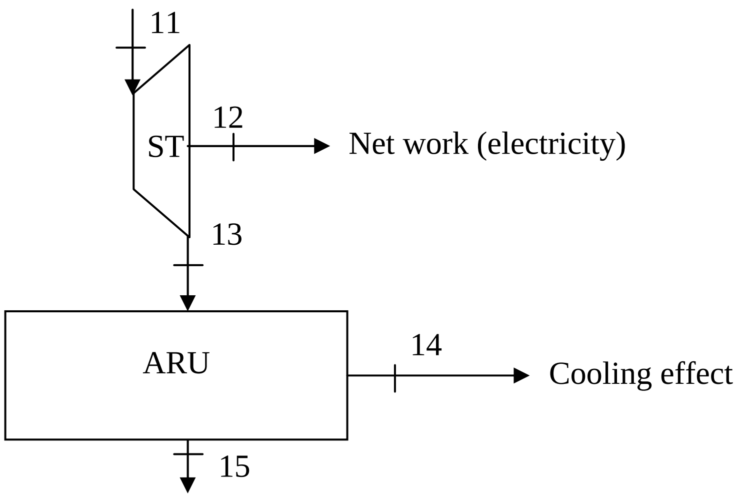

Figure C.1: Steam cycle part of the system adopted

For the case of compressor pressure ratio of 10 and 10 MW net power production in the gas turbine cycle, the cooling effect obtained from the ARU of the adopted system is:

$$\dot{Q}_{REF} = 4835.8 \text{ kW}$$

Revenue obtained from the cooling effect is;

$$\dot{C}_{REF} = 4835.8 \times 7000 \times c_{Ref}$$

$$= 33.85x10^6 \times c_{Ref}$$

The electricity production of the steam cycle for the same parameters is 1741.2 kW (Table 5.2).

Therefore the annual revenue obtained from electricity is:

$$\dot{C}_E = (10000 + 1741.2) \times 7000 \times c_E$$

$$= 82.19x10^6 \times c_E$$

The annual fuel cost is:

$$\dot{C}_F = 28026 \text{ x } 7000 \text{ x } 3600 \text{ x } 5.7 \text{ x } 10^{-6}$$

$$= 4.026 \text{ x } 10^6 \text{ \$/year}$$

The annual investment cost of the system consisting of the gas turbine cycle, steam turbine cycle and the ARU is:

$$\dot{Z} = 0.1 \ (7\,000\,000 + 1740\,000 + 3870\,000)$$

$$= 1\,261\,000 \text{ \$/year}$$

where 0.1 is the capital recovery factor.

The cost balance equation can be written as:

$$\dot{C}_{REF} + \dot{C}_E = \dot{C}_F + \dot{Z} \qquad \textbf{(C.1)}$$

Substituting :

$$33.85x10^6 \times c_{Ref} + 82.19 \times 10^6 \times c_E = 4.026 \text{ x } 10^6 + 1.26 \text{ x } 10^6$$

Simplifying:

$$33.85\ c_{Ref} + 82.19\ c_E = 5.286$$

Considering that approximately 2.5 units of cooling effect can be obtained with 1 unit of work with conventional means, one can write,

$$c_E = 2.5 \times c_{Ref}$$

Solving the above equations yields:

$c_E = 0.0552\ \$/kWh$

$c_{Ref} = 0.022\ \$/kWh$

Case (b)

This case considers using only heat recovery steam generator without the steam cycle.

Figure C.2: Schematic diagram of the heat recovery steam generator of the gas turbine cycle.

Fuel energy input for 10 MW net power production from the gas turbine is 28.026 MW.

Heat that can be recovered from the HRSG is:

$$\dot{Q}_{HRSG} = \dot{m} \times c_p \times (T_6 - T_7)$$

$$= 30.46 \text{ x } 1.147\ (764.6 - 427)$$

$$= 11794.9 \text{ kW.}$$

Heat obtained is multiplied with 0.6 which is the COP_{ARU} to get the cooling effect.

$$\dot{Q}_{REF} = 11794.9 \text{ x } 0.6 = 7076.9 \text{ kW}$$

Revenue obtained from the cooling effect is:

$$\dot{C}_{REF} = 7076.9 \times 7000 \times c_{Ref}$$

where c_{Ref} is the cost of the cooling effect in \$/kWh

The capital cost of the gas turbine cycle including the HRSG is 7 000 000 + 3 870 000 = 10 870 000 $.

Multiplying with the capital recovery factor of 0.1, the yearly investment cost $\dot{Z}$ is found as 1 087 000 $/year.

The annual fuel cost is:

$$\dot{C}_F = 28026 \times 7000 \times 3600 \times 5.7 \times 10^{-6}$$

$$= 4.026 \text{ x } 10^6 \text{ \$/year}$$

where 5.7 x 10^{-6} is the price of fuel in $/kJ and number of hours of operation per year is taken as 7000 h.

Revenue obtained from electricity is:

$$\dot{C}_E = 10000 \times 7000 \times c_E$$

The cost balance equation can be written as:

$$\dot{C}_{REF} + \dot{C}_E = \dot{C}_F + \dot{Z} \quad \textbf{(C.2)}$$

Substituting:

$$49.54 \times 10^6 \times c_{Ref} + 70 \times 10^6 \times c_E = 4.026 \times 10^6 + 1.087 \times 10^6$$

Simplifying

$$49.54 \times c_{Ref} + 70 \times c_E = 5.347$$

Considering that approximately 2.5 units of cooling effect can be obtained with 1 unit of work with conventional means, one can write:

$$c_E = 2.5 \text{ x } c_{Ref}$$

Solving the above equations yields:

$c_E = 0.0595$ \$/kWh.

$c_{Ref} = 0.0238$ \$/kWh.

Case (c)

This case considers the determination of the cost of cooling effect from the data given in the literature.

In the system outlined by Colonna and Gabrielli (2003),

Annual electricity production is:

$$\dot{E}_E = 70.97 \times 10^6 \quad \text{kWh/year}$$

Annual cooling effect is:

$$\dot{Q}_{REF} = 67 \times 10^6 \quad \text{kWh/year}$$

Annual fuel consumption is:

$$\dot{E}_F = 827.6 \times 10^9 \quad \text{kJ/year}$$

Annual capital investment is:

$$\dot{Z} = 1\,269\,156 \quad \text{\$/year}$$

Writing the cost balance equation:

$$\dot{C}_{REF} + \dot{C}_E = \dot{C}_F + \dot{Z} \qquad \textbf{(C.3)}$$

Substituting and simplifying:

$$70.97 \times 10^6 \; c_E + 67 \times 10^6 \; c_{Ref} = 827.6 \times 10^9 \times 5.7 \times 10^{-6} + 1\,269\,156$$

$$70.97 \times c_E + 67 \times c_{Ref} = 5.9865$$

Assuming $c_E = 2.5\ c_{Ref}$ as before; one obtains:

$c_E = 0.06123 \ \$/kWh$

$c_{Ref} = 0.0245 \ \$/kWh$

2. Example calculation of the pay back period

The pay back period for 10 MW net power production in the gas turbine, for a compressor pressure ratio of 10 and natural gas cost of 0.2 \$ / m^3 is given below:

The investment costs for the gas turbine cycle, steam turbine cycle and the ARU have been taken as 7 million \$, 1.74 millon \$ and 3.87 million \$ respectively.

The cost per unit exergy for electricity from the gas turbine and the steam turbine were calculated as 0.041 \$/kWh and 0.083 \$/kWh respectively (Table 5.8). Similarly the cost of cooling effect is 0.1153 \$/kWh. The fuel cost per unit exergy is 0.021 \$/kWh.

Assuming that yearly operating hours is 8400 h, the yearly costs and revenues are found as follows:

Revenue for electricity,

(10000 x 0.041 + 1741.2 x 0.083) 8400 = 4 658 965 \$.

Revenue for cooling effect,

1725.7 x 8400 x 0.1153 = 1 671 968 \$.

Cost of fuel,

28026 x 8400 x 0.021 = 4 943 786 \$.

Net annual operating revenue is thus:

4 658 965 + 671 968 – 4 943 786 = 1 386 146 \$.

The total investment cost is the sum of the costs for the gas turbine cycle, steam turbine cycle and ARU:

7 000 000 + 1 740 000 + 3 870 000 = 12 610 000 \$.

Letting the salvage value to be 10 % of the initial investment, the payback period, pp is found as:

$$pp = \frac{1261000 - 1261000}{1386146} = 8.2 \quad \text{years}$$

CURRICULUM VITAE

Abd Elmonim Mohamed Elamin Elhanan was born in Omdurman, Sudan in March 1950. After primary, intermediate and high school in Omdurman, he attended the Mechanical Engineering Department of the Khartoum Polytechnic Institute where he received his B.Sc. degree in 1974. He entered the University of Leeds in England in 1987 and obtained his M.Sc. degree in combustion and energy in 1989.
He has been pursuing his Ph.D. studies in the Department of Mechanical Engineering of the Istanbul Technical University with a scholarship given by the Sudanese Ministry of Higher Education and Scientific Research since 1998.
He is married and has four children. He speaks Arabic and English.

Fortran File

The thermodynamic analysis program

```
C      ******    ABMT06.FOR   ************************************
C      THIS PROGRAM DOES THE ANALYSIS OF A GAS TURBINE
C      TRIGENERATION SYSTEM. SYSTEM HAS BEEN MODIFIED
C      ON AUGUST 23, 2005.
C      METHODOLOGY IS SIMILAR TO THAT GIVEN IN BEJAN,
C      TSATSARONIS AND MORAN. TABLES ON p. 520 OF THIS
C      REFERENCE ARE USED FOR H, S AND CP CALCULATIONS.
C      c A.Moneim El HANNAN, Taner DERBENTLI, August 23, 2005
C                       Revised  August 27, 2005
C                       Revised  Sept.  19, 2005
C     ************************************************************
       CHARACTER*6 BASLIK(10)
C      PRESSURE, TEMPERATURE, ENTHALPY, ENTROPY
C     ****************************************
       DIMENSION P(17),T(17),H(17),S(17)
C      EXERGIES AT VARIOUS STATES OF THE TRIGENERATION CYCLE
C      EWPHX : SPECIFIC PH. EXERGY OF WATER AT VARIOUS STATES
C      OF THE STEAM CYCLE.
C      ******************************************************
       DIMENSION EX(17),EXPH(17),EXCH(17),EXPHR(17),EWPHX(4)
       DIMENSION EMDOT(17),NSUBA(4),YA(4),NSUBP(5),HPROD(5)
C     YA, YP : MOLE FRACTION OF AIR AND PRODUCTS OF COMBUSTION
C      FOR THE PROD. ORDER IS : CO2, CO, H2O, O2, N2, H2OL, CH4
C      **********************************************************
       DIMENSION YP(7),YPNEW(7),YPS(7),CEX(7)
       DATA PO,TO/101.3,298.15/
       DATA YA/0.0003,0.019,0.2059,0.7748/
       DATA NSUBA/1,3,4,5/
       DATA NSUBP/1,2,4,3,5/
       DATA CEX/14176.,269412.,3951.,8636.,639.,45.,824348./
C      MOLECULAR WEIGHTS
C      ******************
       DATA EMAIR,EMPROD,EMFUEL/28.649,28.254,16.043/
C      CP VALUE OF THE COMB. PRODUCTS AND THE ENTHALPY
C      DIFFERNECES RELATED TO THE VAPOR CYCLE.
C      ************************************************
       DATA CPG,TVAP,DH1W,DH2W,DH3W/1.147,250.4,291.1,
     -1714.1,521.2/
       DATA EWPHX/67.88,69.15,1135.64,623.13/
C      QARU : HEAT TRANSFERRED TO THE ARU PER UNIT MASS OF
C      WATER IN THE STEAM CYCLE, WSTUR, WSPMP : SPECIFIC
C      WORKS OF TURBINE AND PUMP IN THE STEAM CYCLE,
```

```
C      COPVC : ASSUMED COP OF THE VAPOR COMP. REF. CYCLE
C      ****************************************************
        DATA QARU,WSTUR,WSPMP,COPVC/2077.23,449.12,4.67,2.8/
C      INPUT FILE : GTARIN, OUTPUT FILES : AMEXER, GTAROUT
       OPEN(5,FILE='GTARIN.DAT',STATUS='OLD')
       OPEN(6,FILE='GTAROUT.DAT',STATUS='NEW')
       OPEN(8,FILE='AMEXER.DAT',STATUS='NEW')
C      FOLLOWING ARE GIVEN PRESSURES AND TEMPERATURES AT
C      VARIOUS STATE POINTS.
C      *************************************************
      P(9)=0.
      P(10)=0.
      T(9)=0.
      T(10)=0.
      P(11)=4000.
      T(11)=623.
      P(12)=0.
      T(12)=0.
      P(13)=300.
      T(13)=406.6
      P(14)=0.
      T(14)=0.
      P(15)=300.
      T(15)=406.6
      P(16)=4000.
      T(16)=407.6
      P(17)=0.
      T(17)=0.
C      ***************************************************
C      READING INPUT VALUES FROM GTARIN.DAT
C      PRCOMP : PRESSURE RATIO OF THE COMPRESSOR
C      POWNET : NET POWER OF THE CYCLE
C      TCCIN, TCCOUT : COMB. CHAMBER INLET AND EXIT TEMPS.
C      DPAPA, DPAPB : PRESSURE DROPS (%) ON THE AIR SIDE
C      AND THE GAS SIDE OF THE AIR PREHEATER RESP.
C      DPCC,DPHRSG : PRESSURE DROPS IN THE COMB. CHAMBER
C      AND THE HEAT RECOVERY STEAM GENERATOR RESP.
C      ETAC,ETAT,ETAAP, ETAWHB  : EFFICIENCY OF DEVICES
C      YCC : RATIO OF CARBON IN THE FUEL CONVERTED TO CO2
C      TMINC : EXIT TEMP. OF GASES FROM HRSG (ASSUMED 154 C)
C      COPARU : COP OF THE ABSORBTION REF. UNIT
C      RINT : INTEREST RATE, EN : ECONOMIC LIFE OF SYSTEM
C      SPC... : COST PER KW OF DEVICE, ZF : FUEL COST ($/M3)
C      ****************************************************
       READ(5,100)BASLIK
  100  FORMAT(10A6)
      READ(5,110)P1,T1,PRCOMP,POWNET
   10 FORMAT(F8.1,F8.2,F6.1,F10.1)
      READ(5,120)TCCIN,TCCOUT
  120 FORMAT(2F8.1)
```

```
      READ(5,130)DPAPA,DPCC,DPAPB,DPHRSG
  130 FORMAT(4F6.2)
      READ(5,140)PFUEL,TFUEL
  140 FORMAT(F8.1,F8.2)
      READ(5,150)ETAC,ETAT,ETAAP,ETAWHB
  150 FORMAT(4F6.2)
      READ(5,160)YCC,TMINC,COPARU
  160 FORMAT(F6.3,F8.1,F6.2)
      READ(5,170)RINT,EN,SPCGT,SPCARU,SPCST,ZFUEL
  170 FORMAT(F8.3,F5.1,3F8.1,F8.3)
C     **********************************************
C      CALCULATION OF H,S AND E OF AIR AT T(I),P(I)
C      SUBROUTINE PROPER IS USED FOR THIS PURPOSE.
C      **********************************************
       EPS=0.00001
       P(8)=PFUEL
       T(8)=TFUEL
       BEYCC=1.0-YCC
C      ******************************
C      STATE 1 IS INLET TO COMPRESSOR
C      ******************************
       HONE=0.0
       SONE=0.0
       T(1)=T1
       P(1)=P1
       DO 10 I=1,4
       PARP=PO*YA(I)
       NV=NSUBA(I)
       TV=TO
       CALL PROPER(TV,PARP,CPIV,HIV,SIV,NV)
       HONE=HONE+YA(I)*HIV
       SONE=SONE+YA(I)*SIV
   10  CONTINUE
       H(1)=HONE
       S(1)=SONE
       EXPH(1)=0.0
       T1A=T(1)
       TAVE=T1A+100.
       T2SOLD=TAVE
C      *****************************************
C      STATE 2 IS THE EXIT OF THE COMPRESSOR.
C      TEMP AT THE EXIT IS FOUND ITERATIVELY
C      BY CONSIDERING CP AS A FUNCTION OF T.
C      *****************************************
   22  CONTINUE
       Y=(TAVE-273.15)/100.
       CPAV=-0.2*Y**2+1.56*Y+28.48
       CVAV=CPAV-8.314
       AK=CPAV/CVAV
       US=(AK-1.)/AK
```

```
      T2S=T1A*PRCOMP**US
      DELTA=ABS(T2S-T2SOLD)
      IF(DELTA.LT.0.1)GO TO 24
      T2SOLD=T2S
      TAVE=0.5*(T1A+T2S)
      GO TO 22
   24 T(2)=T1A+(T2S-T1A)/ETAC
      P(2)=P(1)*PRCOMP
      HTWO=0.0
      STWO=0.0
      TTWO=T(2)
      DO 26 I=1,4
      NV=NSUBA(I)
      PARP=P(2)*YA(I)
      CALL PROPER(TTWO,PARP,CPIV,HIV,SIV,NV)
      HTWO=HTWO+HIV*YA(I)
      STWO=STWO+SIV*YA(I)
   26 CONTINUE
      H(2)=HTWO
      S(2)=STWO
      EXPH(2)=H(2)-H(1)-TO*(S(2)-S(1))
      WC=H(2)-H(1)
C     **************************************************
C     STATE 3 IS THE EXIT OF THE AIR PREHEATER OR THE
C     INLET TO THE COMB. CHAM., TEMP. TCCIN IS GIVEN
C     **************************************************
      P(3)=P(2)*(1.0-DPAPA)
      T(3)=TCCIN
      HTHRE=0.0
      STHRE=0.0
      TTHRE=T(3)
      DO 34 I=1,4
      NV=NSUBA(I)
      PARP=P(3)*YA(I)
      CALL PROPER(TTHRE,PARP,CPIV,HIV,SIV,NV)
      HTHRE=HTHRE+HIV*YA(I)
      STHRE=STHRE+SIV*YA(I)
   34 CONTINUE
      H(3)=HTHRE
      S(3)=STHRE
      EXPH(3)=H(3)-H(1)-TO*(S(3)-S(1))
C     ****************************************************
C     STATE 4 IS THE EXIT OF COMB. CHAM., TCCOUT IS GIVEN
C     CALCULATIONS IN THE COMBUSTION CHAMBER INVOLVE THE
C     DETERMINATION OF LAMBDA, THE FUEL/AIR RATIO, THEN
C     EXERGY IS CALCULATED TAKING INTO ACCOUNT THE
C     PROBABLE CONDENSATION OF WATER IN THE PRODUCTS WHEN
C     BROUGHT TO ENVIRONMENTAL CONDITIONS.
C     ****************************************************
      TFOUR=TCCOUT
```

```
      T(4)=TCCOUT
      P(4)=P(3)*(1.0-DPCC)
      PFOUR=P(4)
      DO 36 I=1,5
      NV=NSUBP(I)
      CALL PROPER(TFOUR,100.,CPIV,HIV,SIV,NV)
      HPROD(I)=HIV
36    CONTINUE
      HTERA=HTHRE
      TERX=YCC*HPROD(1)+BEYCC*HPROD(2)
      HTERB=0.0003*TERX
      HTERC=0.5*(0.4121-0.0003*YCC)*HPROD(3)
     -+0.019*HPROD(4)+0.7748*HPROD(5)
      HTERD=TERX-0.5*(3.+YCC)*HPROD(3)+2.0*HPROD(4)
      CALL PROPER(TFUEL,PFUEL,CPIV,HIV,SIV,7)
      H(8)=HIV
      S(8)=SIV
      CALL PROPER(TO,PO,CPIV,HIV,SIV,7)
      EXPH(8)=(H(8)-HIV)-TO*(S(8)-SIV)
      ALAM=(HTERB+HTERC-HTERA)/(H(8)-16047.2-HTERD)
      ENPT=ALAM*(1.5-0.5*YCC)-0.00015*YCC+1.00015
      YP(1)=(ALAM+0.0003)*YCC/ENPT
      YP(2)=(ALAM+0.0003)*BEYCC/ENPT
      YP(3)=0.5*(0.4121-ALAM*(3.0+YCC)-0.0003*YCC)/ENPT
      YP(4)=(2.0*ALAM+0.019)/ENPT
      YP(5)=0.7748/ENPT
      DO 38 I=1,5
      YPS(I)=YP(I)
38    CONTINUE
      HFOUR=0.0
      SFOUR=0.0
      DO 42 I=1,5
      IF(YP(I).LT.EPS)GO TO 42
      PARP=YP(I)*PFOUR
      NV=NSUBP(I)
      CALL PROPER(TFOUR,PARP,CPIV,HIV,SIV,NV)
      HFOUR=HFOUR+YP(I)*HIV
      SFOUR=SFOUR+YP(I)*SIV
42    CONTINUE
      H(4)=HFOUR
      S(4)=SFOUR
      PVAP=PO*YP(4)
      IF(PVAP.LT.3.17)GO TO 48
      XTH2O=YP(4)
      XND=1.0-XTH2O
      XGH2O=0.0323*XND
      TOTNG=XND+XGH2O
      DO 44 I=1,5
      YPNEW(I)=YP(I)/TOTNG
44    CONTINUE
```

```
      YP(6)=YP(4)-XGH2O
      YP(4)=XGH2O
      YPNEW(4)=XGH2O/TOTNG
      GO TO 56
   48 CONTINUE
      DO 50 I=1,5
      YPNEW(I)=YP(I)
   50 CONTINUE
   56 HPRODO=0.0
      SPRODO=0.0
C     ****************************************
C     CALCULATION OF THE CHEMICAL EXERGY FOR
C     THE COMBUSTION PRODUCTS
C     ****************************************
      CHEMA=0.0
      CHEMB=0.0
      DO 58 I=1,5
      IF(YPNEW(I).LT.EPS)GO TO 58
      CHEMA=CHEMA+YPNEW(I)*CEX(I)
      CHEMB=CHEMB+YPNEW(I)*ALOG(YPNEW(I))
   58 CONTINUE
      CHEMEX=TOTNG*(CHEMA+8.314*298.15*CHEMB)+YP(6)*CEX(6)
      DO 60 I=1,5
      PERMOL=YPNEW(I)
      NV=NSUBP(I)
      IF(PERMOL.LT.EPS)GO TO 60
      PARP=PERMOL*PO
      CALL PROPER(TO,PARP,CPIV,HIV,SIV,NV)
      HPRODO=HPRODO+YP(I)*HIV
      SPRODO=SPRODO+YP(I)*SIV
   60 CONTINUE
      HPRODO=HPRODO+YP(6)*(-285829.0)
      SPRODO=SPRODO+YP(6)*69.948
      EXPH(4)=(H(4)-HPRODO)-TO*(S(4)-SPRODO)
      P(7)=PO
      P(6)=P(7)/(1.0-DPHRSG)
      P(5)=P(6)/(1.0-DPAPB)
      PRTINV=P(5)/P(4)
      TAVE=T(4)-100.
      T5SOLD=TAVE
C     ************************************************
C     STATE 5 IS THE EXIT OF THE TURBINE
C     TEMP. AT STATE 5 IS CALCULATED ITERATIVELY BY
C     CONSIDERING CP AS A FUNCTION OF T
C     ************************************************
   62 CONTINUE
      CPAV=0.00355*TAVE+30.818
      CVAV=CPAV-8.314
      AK=CPAV/CVAV
      US=(AK-1.)/AK
```

```
      T5S=T(4)*PRTINV**US
      DELTA=ABS(T5S-T5SOLD)
      IF(DELTA.LT.0.1)GO TO 64
      T5SOLD=T5S
      TAVE=0.5*(T(4)+T5S)
      GO TO 62
   64 T(5)=T(4)-ETAT*(T(4)-T5S)
      TFIVE=T(5)
      PFIVE=P(5)
      HFIVE=0.
      SFIVE=0.
      DO 68 I=1,5
      IF(YPS(I).LT.EPS)GO TO 68
      NV=NSUBP(I)
      PARP=YPS(I)*PFIVE
      CALL PROPER(TFIVE,PARP,CPIV,HIV,SIV,NV)
      HFIVE=HFIVE+HIV*YPS(I)
      SFIVE=SFIVE+SIV*YPS(I)
   68 CONTINUE
      H(5)=HFIVE
      S(5)=SFIVE
      EXPH(5)=H(5)-HPRODO-TO*(S(5)-SPRODO)
      ALAMP1=ALAM+1.
      WT=H(4)-H(5)
      ENDAIR=POWNET/(ALAMP1*WT-WC)
      ENDPRD=ENDAIR*ALAMP1
C     **************************************************
C     CALCULATION OF H6 AND T6 AT EXIT OF AIR PREHEATER
C     **************************************************
      HSIXA=H(5)-ENDAIR*(H(3)-H(2))/(ENDPRD*ETAAP)
      PSIX=P(6)
C     **************************************************
C     34.91 IS THE CP OF PRODUCTS AT 875 C IN KJ/KMOL-K
C     **************************************************
      T6P=T(5)-(H(5)-HSIXA)/34.91
   70 HSIX=0.
      SSIX=0.
      DO 72 I=1,5
      IF(YPS(I).LT.EPS)GO TO 72
      NV=NSUBP(I)
      PARP=YPS(I)*PSIX
      CALL PROPER(T6P,PARP,CPIV,HIV,SIV,NV)
      HSIX=HSIX+HIV*YPS(I)
      SSIX=SSIX+SIV*YPS(I)
      CPIX=CPIX+CPIV*YPS(I)
   72 CONTINUE
      DIFREN=HSIXA-HSIX
      IF(ABS(DIFREN).LT.1.0)GO TO 76
      T6P=T6P+(HSIXA-HSIX)/CPIX
      GO TO 70
```

```
 76   H(6)=HSIX
      S(6)=SSIX
      T(6)=T6P
      EXPH(6)=H(6)-HPRODO-TO*(S(6)-SPRODO)
C     ***********************************************
C     CALCULATION OF H7,S7 AND EX7, EXIT OF HRSG
C     A MINIMUM TEMPERATURE IS SITIPULATED TO BE ON
C     THE SAFE SIDE FOR DANGER OF CONDENSATION.
C     THIS TEMPERATURE IS APPROX. 427 K  (154 C).
C     ***********************************************
      TSEVEN=TMINC
      PSEVEN=P(7)
      HSEVEN=0.
      SSEVEN=0.
      DO 82 I=1,5
      IF(YPS(I).LT.EPS)GO TO 82
      NV=NSUBP(I)
      PARP=YPS(I)*PSEVEN
      CALL PROPER(TSEVEN,PARP,CPIV,HIV,SIV,NV)
      HSEVEN=HSEVEN+HIV*YPS(I)
      SSEVEN=SSEVEN+SIV*YPS(I)
 82   CONTINUE
      T(7)=TSEVEN
      H(7)=HSEVEN
      S(7)=SSEVEN
      EXPH(7)=H(7)-HPRODO-TO*(S(7)-SPRODO)
C     ***************************************************
C     STATE 8 REPRESENTS THE ENTRY OF FUEL
C     STATES 9 AND 10 ARE FICTITIOUS STATES REPRESENTING
C     WORK FLOWS (TO COMP. AND NET) FROM THE GAS TURBINE
C     ***************************************************
      EMDOT(1)=ENDAIR*EMAIR
      EMDOT(2)=EMDOT(1)
      EMDOT(3)=EMDOT(1)
      EMDOT(4)=ENDPRD*EMPROD
      EMDOT(5)=EMDOT(4)
      EMDOT(6)=EMDOT(4)
      EMDOT(7)=EMDOT(4)
      EMDOT(8)=ALAM*EMFUEL*ENDAIR
      EMDOTF=EMDOT(8)
      POWERC=WC*ENDAIR
      POWERT=WT*ENDPRD
      EXPHR(1)=EXPH(1)*ENDAIR
      EXPHR(2)=EXPH(2)*ENDAIR
      EXPHR(3)=EXPH(3)*ENDAIR
      EXPHR(4)=ENDPRD*(EXPH(4)+CHEMEX)
      EXPHR(5)=ENDPRD*(EXPH(5)+CHEMEX)
      EXPHR(6)=ENDPRD*(EXPH(6)+CHEMEX)
      EXPHR(7)=ENDPRD*(EXPH(7)+CHEMEX)
      EXPHR(8)=ALAM*ENDAIR*(EXPH(8)+CEX(7))
```

```
      PWRNET=POWERT-POWERC
      EXPHR(9)=PWRNET
      EXPHR(10)=POWERC
      COSTGT=SPCGT*PWRNET
C     ************************************************
C     CALCULATION OF THE STEAM SIDE AND ARU EXERGIES
C     STATES 11 AND 13 ARE ENTRY AND EXIT OF THE S.T.
C     STATES 15 AND 16 ARE ENTRY AND EXIT OF PUMP
C     ENTRY AND EXIT TO ARU HX ARE STATES 13 AND 15
C     FICTITIOUS STATES 12 AND 17 REPRESENT WORK OF
C     THE STEAM TURBINE AND THE PUMP.
C     FICTITIOUS STATE 14 REPRESENT THE COOLING
C     EFFECT OF THE ARU.
C     ************************************************
      CAPRAT=CPG*EMDOT(6)
      TPINCH=TVAP+20.0
      EMW=CAPRAT*(TPINCH-TMINC+273.)/DH3W
   84 TMT=TPINCH+EMW*(DH2W+DH1W)/CAPRAT
      TFARK=T(6)-273.0-TMT
      IF(ABS(TFARK).LT.0.1)GO TO 86
      EMW=EMW+0.01*TFARK
      GO TO 84
   86 EMDOT(11)=EMW
      EMDOT(13)=EMW
      EMDOT(15)=EMW
      EMDOT(16)=EMW
      EMDOT(17)=EMW
      EXPHR(11)=EMW*EWPHX(3)
      EXPHR(12)=EMW*WSTUR
      EXPHR(13)=EMW*EWPHX(4)
      ARUCE=EMW*QARU*COPARU
      EX14TR=ARUCE/COPVC
      EXPHR(14)=ARUCE/COPVC
      EXPHR(15)=EMW*EWPHX(1)
      EXPHR(16)=EMW*EWPHX(2)
      EXPHR(17)=EMW*WSPMP
      COSTST=SPCST*EXPHR(12)
      COSTARU=SPCARU*ARUCE
C     ************************************************
C     END OF CALCULATIONS OUTPUT IS WRITTEN TO FILES :
C     8 = AMEXER.DAT AND 6 = GTAROUT.DAT
C     ************************************************
      WRITE(8,400)BASLIK
  400 FORMAT(10A6)
      WRITE(8,410)RINT,EN
  410 FORMAT(F10.3,F5.1)
      WRITE(8,420)COSTGT,COSTARU,COSTST
  420 FORMAT(3E10.3)
      WRITE(8,430)EMDOTF,ZFUEL
  430 FORMAT(2F10.3)
```

```
      WRITE(8,440)EXPHR
 440  FORMAT(6F10.1)
C     ********************************************
      WRITE(6,300)BASLIK
 300  FORMAT(8X,10A6,/,
     -8X,'c (2005) A. Elhannan, T. Derbentli',/)
      WRITE(6,310)P1,T1
 310  FORMAT(8X,'AIR INLET  PRES. : ',F8.1,' kPa, AND TEMP. :',
     -F7.1,' K')
      WRITE(6,312)PFUEL,TFUEL
 312  FORMAT(8X,'FUEL INLET PRES. : ',F8.1,' kPa, AND TEMP. :',
     -F7.1,' K')
      WRITE(6,314)PRCOMP,POWNET
 314  FORMAT(8X,'COMP. PRES. RATIO :',F6.1,' ,    NET POWER : ',
     -F8.1,' kW')
      WRITE(6,320)TCCIN,TCCOUT
 320  FORMAT(8X,'COMB. CHAMBER INLET AND OUTLET TEMPS : ',
     -F6.1,F7.1,' K',/)
     WRITE(6,322)DPAPA,DPAPB,DPCC
 322  FORMAT(8X,'PRESSURE DROPS AS FRACTION OF INLET PRESSURE,
     -/8X,'AIR PREHEATER, AIR SIDE : ',F4.2,'  GAS SIDE : ',F4.2,
     -/8X,'COMBUSTION CHAMBER : ',F4.2,/)
      WRITE(6,326)ETAC,ETAT,ETAAP
 326  FORMAT(8X,'COMPRESSOR AND TURBINE EFFICIENCIES :
      ',F4.2,F5.2,/
     -8X,'AIR PREHEATER EFFECTIVENESS         : ',F4.2)
      WRITE(6,330)TMINC
 330  FORMAT(8X,'FLUE GAS TEMPERATURE : ',F6.1,' K')
      WRITE(6,340)COPARU
 340  FORMAT(8X,'COP OF THE ABSORPTION REF. UNIT : ',F5.2,/)
      WRITE(6,344)
 344  FORMAT(8X,'STATE   MDOT(kg/s)  P(kPa)      T(K)',
     -'EX RATE (kW)',/)
      DO 90 I=1,17
      WRITE(6,350)I,EMDOT(I),P(I),T(I),EXPHR(I)
 350  FORMAT(8X,I3,5X,F8.2,2F10.1,F14.1)
  90  CONTINUE
      STOP
      END
C     ****************************************************
      SUBROUTINE PROPER(TEMP,PRES,CPV,HV,SV,MKOD)
C     T (K) , P (kPa) BIRIMLERINDE OLMALIDIR.
C     MKOD : SUBSTANCE CODE = CO2,CO,H2O,O2,N2,H2O(L),CH4
C     ****************************************************
      DIMENSION HARTI(7),SARTI(7),A(7),B(7),C(7),D(7)
      DATA HARTI/-413.886,-120.809,-253.871,-9.589,-9.982,
     --289.932,-81.242/
      DATA SARTI/-87.078,18.937,-11.75,36.116,16.203,
     --67.147,96.731/
      DATA A/51.128,30.962,34.376,29.154,30.418,20.355,11.933/
```

```
DATA B/4.368,2.439,7.841,6.47,2.544,109.198,77.647/
DATA C/-1.469,-0.28,-0.423,-0.184,-0.238,2.033,0.142/
DATA D/3*0.0,-1.017,0.0,0.0,-18.414/
M=MKOD
Y=TEMP/1000.
PPART=PRES
YKAR=Y*Y
YKUB=YKAR*Y
HAV=HARTI(M)
SAV=SARTI(M)
AV=A(M)
BV=B(M)
CV=C(M)
DV=D(M)
CPV=AV+BV*Y+CV/YKAR+DV*YKAR
HV=1000.0*(HAV+AV*Y+0.5*BV*YKAR-CV/Y+DV*YKUB/3.0)
SBO=SAV+AV*ALOG(TEMP)+BV*Y-0.5*CV/YKAR+0.5*DV*YKAR
SV=SBO-8.314*ALOG(PPART/100.0)
RETURN
END
```

Input data for the thermodynamic analysis program is given below:

Table A.1. Input data of the thermodynamic analysis program

ANALYSIS OF THE GAS TURBINE ARU TRIGENERATION SYSTEM					
101.3	298.15	10.0	10000.0		
850.0	1520.0				
0.05	0.05	0.03	0.05		
1200.0	298.15				
0.86	0.86	0.95	1.00		
1.000	427.0	0.60			
0.100	10.0	700.0	800.0	1000.0	0.2

First row shows the inlet pressure of the compressor in kPa, the inlet temperature of the compressor in degree K, the pressure ratio $\frac{P_2}{P_1}$ and the power output of the turbine in kW respectively.

Second row shows combustion chamber inlet temperature degree K, the turbine inlet temperature degree K respectively.

Third row shows the pressure drop of the air preheater on the air side, the pressure drop of the combustion chamber, the pressure drop of the air preheater on the gas side and pressure drop of the heat recovery steam generator respectively.

Fourth row shows the inlet pressure of the fuel (methane) in kPa and the inlet temperature of the fuel in degree K.

Fifth row shows the efficiency of the compressor, the efficiency of the turbine, effectiveness of the air preheater and effectiveness of the heat recovery steam generator respectively.

Sixth row shows the quality x of the steam, the exhaust temperature in degree K and the coefficient of performance of the ARU respectively.

Seventh row shows the interest rate, economic life of the investment, the specific cost of the gas turbine in \$/kW, the specific cost of the ARU in \$/kW, the specific cost of the steam turbine in \$/kW and cost of fuel in \$/m^3 respectively.

Output data for the thermodynamic analysis program is given below:

Table A.2. Output data of the thermodynamic analysis program

STATE	$\dot{m}$ (kg/s)	P (kPa)	T (K)	EX RATE (kW)
1	31.43	101.3	298.1	.0
2	31.43	810.4	565.2	8225.4
3	31.43	769.9	850.0	13836.8
4	32.00	731.4	1520.0	34213.4
5	32.00	109.9	1048.7	14536.2
6	32.00	106.6	774.5	7369.6
7	32.00	101.3	427.0	957.1
8	.57	1200.0	298.1	29269.4
9	.00	.0	.0	10000.0
10	.00	.0	.0	8884.0
11	4.23	4000.0	623.0	4805.9
12	.00	.0	.0	1900.6
13	4.23	300.0	406.6	2637.0
14	.00	.0	.0	1883.7
15	4.23	300.0	406.6	287.3
16	4.23	4000.0	407.6	292.6
17	.00	.0	.0	19.8

First column shows the number of state points of the system.

Second column shows the mass flow rates at different state points of the system in kg/s.

Third column shows the pressures at different state points of the system in kPa.

Fourth column shows the temperatures at different state points of the system in degree K.

Fifth column shows the exergy rates at different state points of the system in kW.