Performance Analysis of Gas Turbine Cogeneration Systems

By

Ashutosh S. Anagal

A Thesis Submitted in Partial Fulfillment
of the Requirements for the Degree of

Master of Applied Science in Mechanical Engineering

Faculty of Engineering and Applied Science

University of Ontario Institute of Technology

June 2014

Abstract

Cogeneration is a highly efficient approach to generating electricity and process heat from the same fuel source. It is an approach of utilizing waste energy products for a useful purpose that significantly improves the optimal provision of the different grades of energy from high-quality, high-grade electricity or power to low quality and low-grade heat. As a result, combined heat and power can be applied to various situations using various technologies.

In the present work, performance of two gas turbine cogeneration systems is analyzed using the first and second laws of thermodynamics. A basic gas turbine cogeneration system and a steam injection gas turbine cogeneration system are investigated. In the analysis the system components, compressor, combustion chamber, turbine and heat recovery steam generator are modeled. The CO_2 emissions of both systems are also evaluated. In this investigation the decrease in CO_2 emissions was found in the steam injected gas turbine cogeneration system as compared to the basic gas turbine cogeneration system.

In the parametric study the influence of pressure ratio and turbine inlet temperature on the performance characteristics of the systems such as specific work, energy and exergy efficiencies are investigated. In the efficiency calculations all the three forms of outputs: power, heat and cogeneration are considered. Thus power generation, heat generation and cogeneration efficiencies are estimated. The carbon dioxide emissions are estimated for both systems. In addition, the effect of pressure ratio and turbine inlet temperature on the CO_2 emissions is studied. The results show that pressure ratio and turbine inlet temperature are the key operating variables to optimize the performance of the cogeneration systems.

The second law analysis revealed that the maximum exergy destruction occurs in the combustion chamber. In this analysis, the chemical and physical component of the exergy is considered.

The role of specific heat as the function of temperature in estimating the performance of cogeneration is also investigated and the performance results are compared with the results assuming constant specific heat. The results with specific heat as the function of temperature are more realistic and accurate compared to the actual performance of the cogeneration systems.

The analysis and the results are useful for optimizing the operating parameters of the gas turbine cogeneration systems and thereby enhancing the system performance. The study is also useful for choosing appropriate cogeneration system for a specific application.

Acknowledgements

I would like to express deepest appreciation to my Supervisor Dr. B. V. Reddy who has been instrumental in the successful completion of this thesis. I feel motivated and encouraged every time I attend his meeting. Without his guidance and persistence help this thesis would not have been possible.

I wish to express my deep sense of gratitude to Dr. T Srinivas for the encouragement and guidance in his brief stay in Canada. He has been my inspiration as I hurdle all the obstacles in the completion of this thesis work.

I owe a great deal of appreciation and gratitude to the Co supervisor Dr. M. Agelin-Chaab for his moral support, guidance and advice in completion of the thesis especially his pain staking effort in proof reading the drafts.

Last but not least, I would like to thank my wife Sneha, my daughters Mrinal and Mrudul for their understanding, patience, sacrifices and wishes in the process of completion of this thesis.

Table of Contents

List of Tables

List of Figures

Nomenclature

u	-	Internal energy (kJ/kg)
c_v	-	Specific heat at constant volume (kJ/kg K)
p	-	Pressure (kPa)
R	-	Universal gas constant (kJ/kg K)
h	-	Specific enthalpy (kJ/kg)
s	-	Specific entropy (kJ/kg .K)
c_p	-	Specific heat at constant pressure (kJ/kg K)
k	-	Specific heat ratio
x_k	-	Fraction of mole of k th component in chemical equation (mole)
N	-	Total number of moles (mole)
n_k	-	Fraction of mole of kth component in chemical equation (mole)
G	-	Gibbs function (kJ/kmole)
$\bar{h}^0$	-	Enthalpy of formation at ambient temperature (kJ/kmole)
$\bar{s}^0$	-	Entropy at ambient temperature (kJ/kmole K)
e	-	exergy (kJ/kg)
$\dot{m}$	-	mass flow rate(kg/s
$\bar{e}^{CH}$	-	Molar chemical exergy (kJ/kmole)
Q_{HRSG}	-	Heat energy in HRSG (kJ)
W_{net}	-	Net work output. (MW)
p_s	-	Steam pressure (k Pa)

h_f - Enthalpy of formation (kJ/kmole)

T_s - Steam temperature (°C)

m_f - Rate of mass flow (fuel) (kg/s)

CV - Calorific value (MJ/kg)

Q_P - Process heat energy (kJ)

Q_s - Process heat demand (kJ)

Q_{rev} - Reversible generated heat (kJ)

c_{pa} - Specific heat of air (kJ/kg K)

$\dot{m}_a$ - Air flow rate (kg/s)

$\dot{m}_f$ - fuel flow rate (kg/s)

$\dot{m}_g$ - Gas flow rate (kg/s)

$\dot{m}_s$ - Steam flow rate (kg/s)

$\eta_{compressor}$ - Isentropic efficiency of compressor

η_{Power} - Power efficiency

η_{cogen} - Cogeneration efficiency

$\eta_{powerexergy}$- Exergy efficiency of power generation

$\eta_{totalexergy}$- Total Exergetic efficiency

ε - Chemical exergy

$\varepsilon_{phstack}$ - Exergy loss at the stack

i_{comp} - Irreversibility in compressor (kJ /kg)

$i_{combustor}$ - Irreversibility in combustor (kJ/kg)

i_{Tur} - Irreversibility in turbine (kJ/kg)

i_{HRSG} - Irreversibility in heat recovery steam generator (kJ/kg)

r_p - Pressure ratio

M - Molecular weight (kg/mole)

$\dot{m}_{sin\ j}$ - Mass of injected steam (kg/s)

$\dot{m}_{gs}$ - Mass of combustion gas in STIG(kg/s)

X - Moles of air in combustion process

c_{pg} - Specific heat of combustion gas (kJ/kg .K)

h_g - Specific enthalpy of combustion gas (kJ/kg)

η_{tur} - Isentropic efficiency of the turbine

η_{Comp} - Isentropic efficiency of the compressor

η_{mech} - Mechanical efficiency of the gas turbine component

$r_{p_{opt}}$ - Optimum pressure ratio

β- - Heat demand ratio

Abbreviations

TIT - - Turbine inlet temperature (K)

Ref- - Reference

Ph- - physical

Ch- - Chemical

Combgas - Combustion gas

Sgen- - Entropy generation

η_{cogen}- - Cogeneration efficiency

GHGs - Green house gases

ppm - Parts per million

Sec - Second

PPT - Pinch point temperature

C - Compressor

CC - Combustion chamber

T - Turbine

HRSG - Heat recovery steam generator

A – Inlet air

F - Inlet fuel

G - Combustion gas

S_p - Process steam

Wp - Water inlet for process steam

ST - Stack outlet

P - Power output

Si – Steam injection outlet

SFC – Specific fuel consumption

BGCS – Basic gas turbine cogeneration system

STIG - Steam injected gas turbine cogeneration system

GT – Gas turbine

GTCS – Gas turbine cogeneration systems

Qph – Process heat

Qmax – Maximum process heat

CHAPTER 1. INTRODUCTION

1.1 Background

Energy services are fundamental in order to achieve sustainable development [1]. The statistics from energy related agencies indicated that the power generation based on fossil fuel are and will have the largest share in global power generation [1]. According to the Energy Information Administration (EIA) the world energy consumption is projected to increase by 50 % from 2005 to 2030[3]. Figure 1. represents the energy demand projections up to 2030[2].

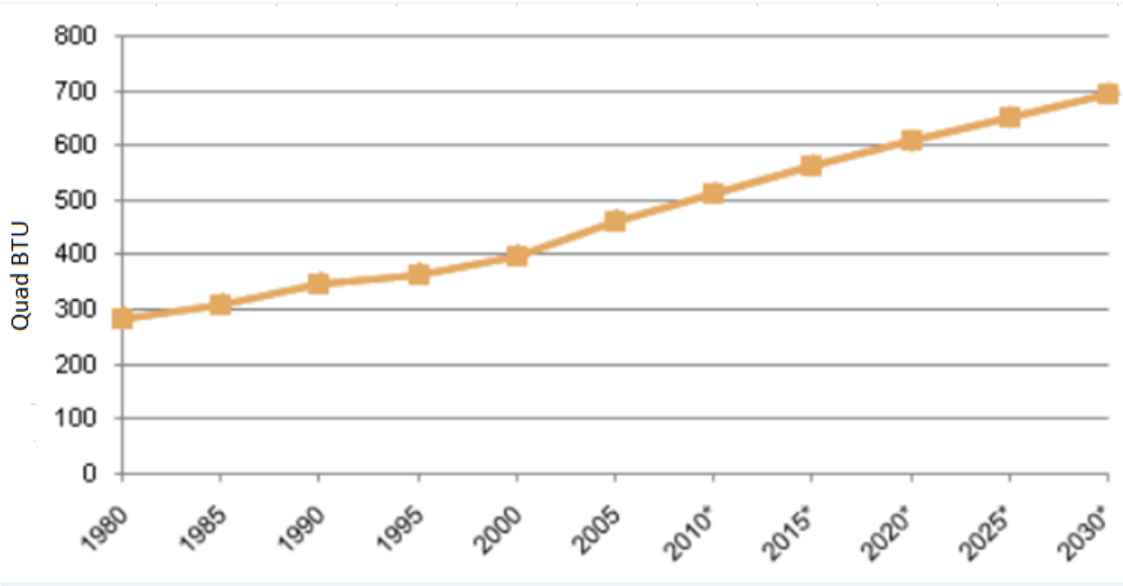


Figure 1.1 World energy demand projections [Source: International energy outlook 2013][3]

The power generation is responsible for 32 % of total fossil fuel consumption and about 41% of total energy related CO2 emissions [2]. Figure 1. shows the forecast about demand for electricity over the period of two decades. In the Annual Energy Outlook report it is forecasted that from 2010 to 2040 the demand for electricity will grow by 2.4 % per year on average and that most of the power generating capacity will be natural gas fired combined cycles [2].

In order to meet future energy demand growth and to lower CO_2 emissions, it is imperative to focus on the energy efficiency improvement initiatives as there is a lot of scope to improve technologies on fossil fuelled power generation and more specifically on improving conversion efficiency, since any gains would result in substantial CO_2 and fuel savings. For example, each percentage point efficiency increase is equivalent to about 2.5 % reduction of CO_2 emissions [2].

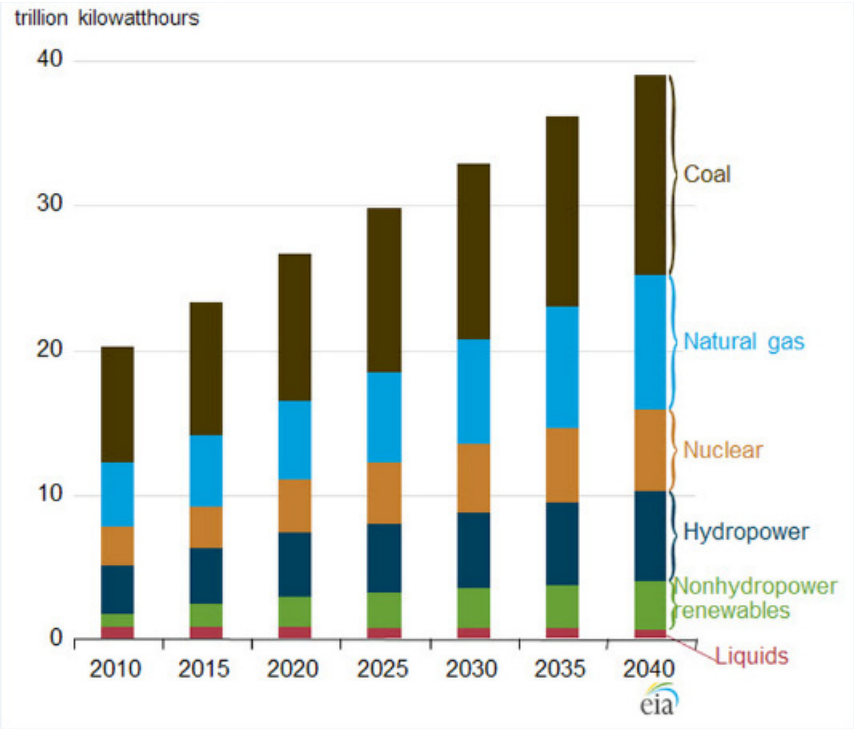


Figure 1.2 Share in Electricity generation by fuel. Source: EIA, International energy outlook 2013[2]

The fossil fuel power generation is popular as the mainstay of supply due to: a) the ability to cope with variability in electricity demand throughout the day and seasonal variability; and b) the reasonable generation costs.

In the current power generation technologies, the heat energy produced during combustion of fossil fuels is utilized to generate power by passing working media steam or gas through a turbine to generate electricity. The current power generation

technologies can be categorized based on the type of fuel used (coal, lignite, oil or natural gas).

1.2 Coal Fired Power Plants

Among the fossil fuels, coal is the most widely used fuel in power plants [4]. Coal plays an important role in global power generation. Coal-fired power plants currently have share of 39-42% of global power generation [3]. Coal, which is readily available in most of the developing and developed world, has been used as a major source of fuel before the industrial revolution, even in ancient human civilizations. Presently three types of coal power plants are commonly used: (a) Pulverized coal-fired (PCF) power plants. (b) Fluidized bed combustion (FBC) plants (c) Integrated gasification combined cycle (IGCC) plants.

1.2.1 Advantages of Coal Power Plant

Coal power plants have been popular since past due to some of its advantages include affordability, reliability, abundance, known technologies, safety, and efficiency.

Energy produced from coal fired plants is relatively inexpensive and more affordable than other energy sources. Another advantage of the coal is that it is abundant. Moreover, it is not expensive to extract and mine from coal deposits. Consequently, its price remains low compared to other fuel and energy sources. There are approximately over 300 years of economic coal deposits still accessible. With this great amount of coal available for use, coal fired plants can be continuously fueled in many years to come. One of the significant advantages of coal fired plants is reliability. Coal's ability to supply power during peak power demand either as base power or as off-peak power is greatly valued as a power plant fuel. It is with this fact that advanced pulverized coal fired power plants are designed to support the grid system in avoiding blackouts. The production and use of coal as a fuel are well understood, and the technology required in producing it is constantly advancing. Moreover, coal-mining techniques are continuously

enhanced to ensure that there is a constant supply of coal for the production of power and energy. And last but not least, generally, coal fired plants are considered safer than nuclear power plants. A coal power plant's failure is certainly not likely to cause catastrophic events such as a nuclear meltdown would. The coal power plants are safer to operate and maintain.

1.2.2 Disadvantages of coal-fired power plants

On the other hand, there are also some significant disadvantages of coal fired plants including Greenhouse Gas (GHG) Emissions, mining destruction, generation of millions of tons of waste, and emission of harmful substances.

Coal leaves behind harmful by-products upon combustion. These by-products cause a lot of pollution and contribute to global warming. The increased carbon emissions brought about by coal fired plants has led to further global warming which results in climate changes. Mining of coal not only results in the destruction of habitat and scenery, but it also displaces humans as well. In many countries where coal is actively mined, many people are displaced in huge numbers due to the pitting of the earth brought about by underground mining. Places near coal mines are unsafe for human habitation as the land could cave in at any time. Millions of tons of waste products which can no longer be reused are generated from coal fired plants. Aside from the fact that these waste products contribute to waste disposal problems, these also contain harmful substances. Thermal plants like coal fired plants emit harmful substances to the environment. These include mercury, sulfur dioxide, carbon monoxide, mercury, selenium, and arsenic. These harmful substances not only cause acid rain but also are very harmful to humans as well.

The coal also has traces of other naturally occurring elements including thorium, uranium and radon. Burning coal concentrates these substances in the fly ash that escapes from the power plant's smokestack. While the amounts of radioactivity are too

small for scientists to consider it a health threat, coal plants put measurably more radiation into their local communities than nuclear plants do.

1.3 Oil Fired Power Plants

Oil is mostly known as crude oil or condensate, but includes all liquid hydrocarbon fossil fuels. Petroleum and liquefied petroleum gas (LPG) are the most common types of fuel obtained from oil extraction and refining. The share of oil in global power generation is quite less compared to coal and gas. In energy supply the oil is mainly used in the transportation sector. Three technologies are used to convert oil into electricity: (a) Conventional steam - Oil is burned to heat water to create steam in order to generate electricity. (b) Combustion turbine - Oil is burned under pressure to produce hot exhaust gases which spin a turbine to generate electricity; and (c) Combined-cycle technology - Oil is first combusted in a combustion turbine, using the heated exhaust gases to generate electricity. After these exhaust gases are recovered, they heat water in a boiler to create steam to drive a second turbine.

1.4 Gas Turbine Power Plants

Gas power plants are increasingly popular because of higher energy conversion efficiencies and low emissions.

1.4.1 Advantages of gas turbine plants

The size and weight of the gas turbine plant are less for large capacities compared to the steam power plant. Water needed to run the gas turbine plant is less compared to the steam plant. Gas turbine plants can be put on load easily and they can be started quickly. The maintenance cost of the gas turbine plant is less. The installation of the gas turbine plant is easier because of the absence of a boiler, evaporator, compensating system, etc. The gas turbine power plant does not require heavy foundations and buildings.

1.4.2 Disadvantages of gas turbine plants

Net output from the gas power plant is low; this is because a major portion of the energy is required to drive the compressor. Temperature of the combustion products is too high. So, even at moderate pressure, more care should be taken.

1.5 Efficiencies of Fossil Fuel Power Plants

The efficiencies of coal-fired plants depend on a range of factors including the technology employed, the type and quality of coal used and the operating conditions and practices. The average efficiency of electricity production from coal in both public electricity-only and public CHP plants is 32%-43% [3-9]

The majority of oil-fired electricity production is in electricity-only plants. The average efficiency of oil-fired electricity production is in the range of 23%-43% [3-9].

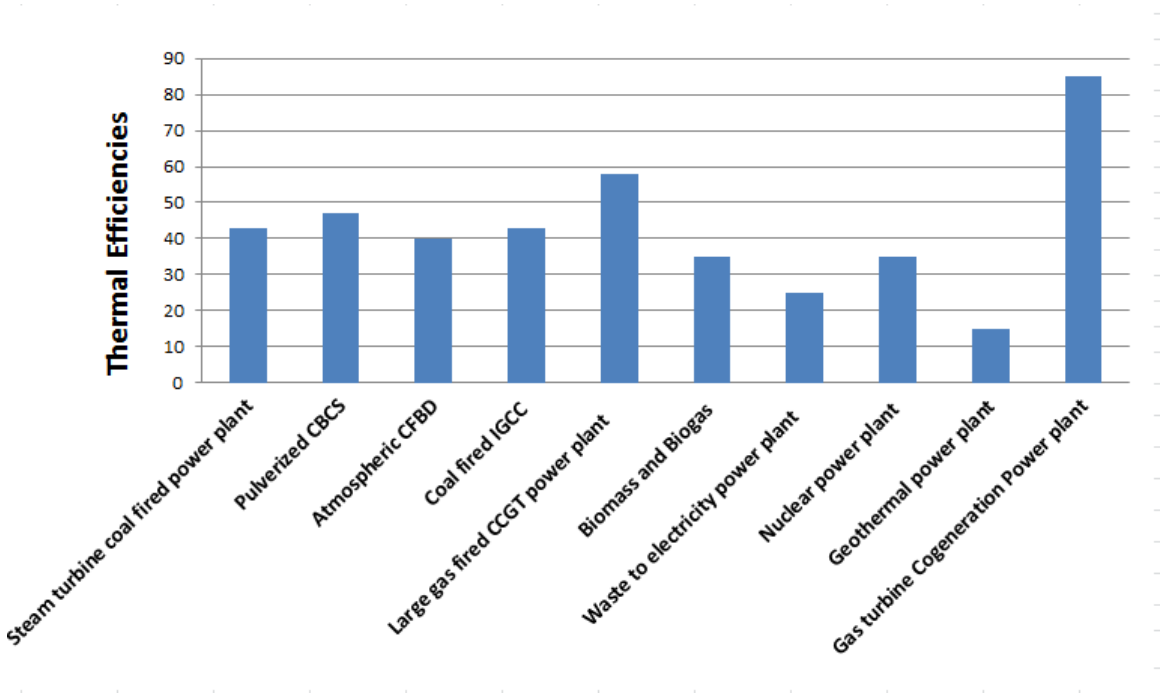


Figure 1.3 Power efficiencies of fossil fuel power plants [4]

The gas turbine plants are more efficient than coal and oil fired power plants. Gas turbine cogeneration technologies such as combined cycle, cogeneration can offer higher efficiencies and generate less CO_2 emissions per unit power output. The average efficiency of natural gas-fired electricity CHP plants (CCGT) is in the range of 31-55% [3-9]. The latest CCGTs can have efficiencies of about 60% [4].

Figure 1.33 summarize the efficiencies of various thermal power systems. Among these thermal power systems, gas turbine cogeneration has the highest potential to achieve the overall thermal efficiency [4].

1.6 Cogeneration (Combined Heat and Power)

As the name implies, cogeneration is the combined production of electrical power and process heat from the same fuel source [6]. By making use of waste from one process in the production of the other more substantial gains in energy efficiency can be realized compared with independent production of both products. This also results in substantial reduction in CO_2 emissions [2]. This is also called combined heat and power.

A cogeneration system comprises a prime mover (heat engine), a power generator, heat recovery steam generator and electrical interconnections.

Figure 1.44 represent a comparison of conventional power generation and combined heat and power. Conventional thermal power plants are inherently inefficient systems. They burn fuel to produce high temperature steam which, under pressure, turns turbines to generate electricity. In the process, about one-third of the energy of the original fuel is converted to high pressure steam to produce electricity. Typically, the remainder of the energy is exhausted as waste heat or discharged as warm water.

By contrast, cogeneration makes use of the waste heat, usually in the form of the steam exhausted from the power generation turbines. This excess steam is suitable for

generating additional electricity or for many industrial or commercial operations that require heat.

As a result of its re-use of an energy source, cogeneration is far more efficient than conventional generation. Cogeneration plants boost power efficiencies in the range of 50 – 70 per cent. The power efficiencies can be as high as 90 per cent over short periods [4].

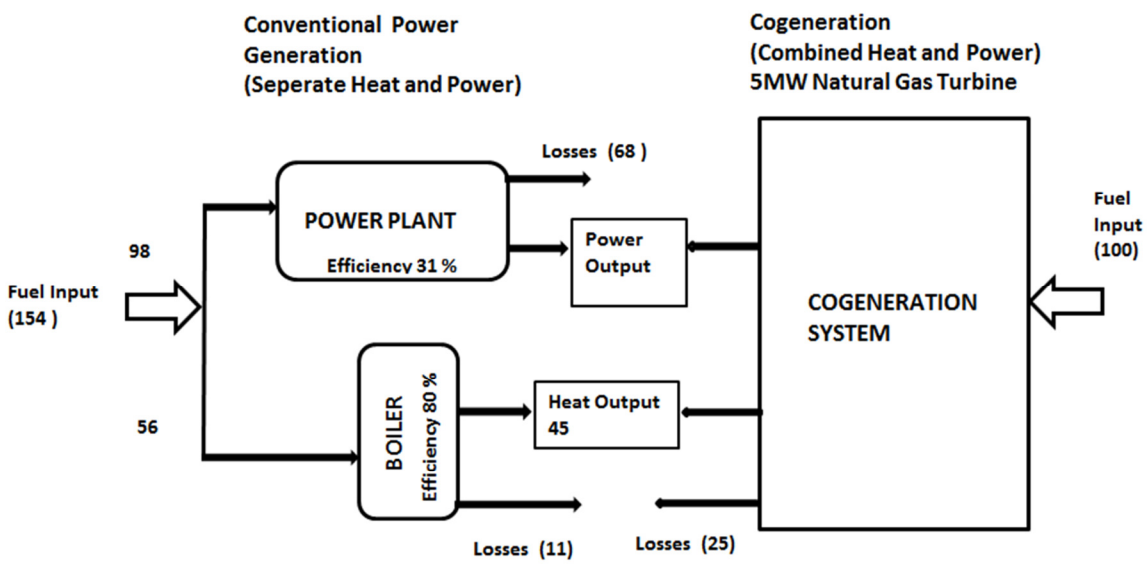


Figure 1.4 Comparison of separate heat & power vs. combined heat & power [6] [Source: U.S. EPA: Combined Heat and power Partnership].

1.6.1 Classification of cogeneration systems based on prime mover

Cogeneration plants are classified by the type of prime mover they use to drive the electrical generator. These include the following.

1.6.1.1 Steam turbine cogeneration plants

In the type of cogeneration plants, the thermal needs of the site determine the heat and power ratio. The lower the grade/quality of the steam required more the amount of power generated per unit of fuel.

The types of steam turbine cogeneration plants are:

a. Back pressure turbine, and

b. Condensing turbines

In the back pressure turbines exhaust steam pressure is always more than the atmospheric pressure however, in condensing turbines; the exhaust steam pressure is always lower than that of the atmosphere (i.e. vacuum). In either case the steam can be extracted at a required pressure. Condensing turbines produce more electricity per unit fuel than back pressure turbines since more energy is extracted by the turbine. As a result there will be less energy available for heating applications. The steam turbine cogeneration plants are available in the range of 50 kW- 250MW [6].

1.6.1.2 Gas turbine cogeneration systems

These types of cogeneration plants produce more electricity per unit of fuel compared to steam turbines. In these systems average heat to power ratio is 2: 1[6]. A provision of supplemental heating through secondary firing to the exhaust gases increases this ratio. The electrical output can be increased by injecting steam into the combustion chamber. The thermal energy from high temperature turbine exhaust gases is extracted in waste heat boiler to meet thermal requirements of the site or other applications. The power in the range of 500kW to 250 MW can be generated by these systems [6]. The gas turbine cogeneration systems are discussed in detail in section 1.7.

1.6.1.3 Combined cycle gas turbine cogeneration systems

These cogeneration systems work on combined cycle. As name suggests, in this system a gas turbine is combined with steam turbine to generate power and heat. Combining gas and steam cycles results in improved thermal efficiency and savings in fuel consumption. The advantage of this system has high electrical efficiency as compared with the rest of the systems.

1.6.1.4 Reciprocating engines cogeneration systems

These are internal combustion engines. Like gas turbines, thermal output in these systems can be increased by supplemental firing.

It is rather difficult to extract heat from reciprocating engines. Since there are two heat sources, one from exhaust gas and the other from engine cooling system, these systems have potential to produce more electrical energy per unit of fuel, more than combined cycle and GT cogeneration systems. The available sizes are in the range of less than 5MW and are particularly used in generator set applications [6].

1.6.1.5 Microturbine based cogeneration systems

A micro turbine is a small, compact system that combines a gas turbine, compressor and generator on a shaft. This low-cost, flexible unit can run on any liquid fuel, including natural gas, flare gas, landfill gas and coal extraction. These are available in the range of 30kW- 250kW [6], while conventional gas turbines are available in the range of 500kW-250MW. Microturbines run at high speeds and, like larger gas turbines, can be used in power-only generation or cogeneration (CHP) systems. Micro turbines are small gas engines. In microturbines the working principle is same as conventional gas turbines where the compressed air from a compressor is preheated in a recuperator using the turbine exhaust. Microturbines operate on the same thermodynamic cycle as conventional gas turbines. Although a general trend in conventional gas turbine technology with regards to performance improvement is to increase higher turbine inlet temperature and pressure ratios, in microturbines the turbine inlet temperature is limited to 982°C (1800°F) and a low pressure ratio is maintained to 3.5-4 [36].

1.6.1.6 Fuel cells based cogeneration systems

In fuel cells chemical energy is converted into the electrical energy. The electrochemical reaction between hydrogen and oxygen produces chemical energy. It has electrolytic

solution to combine hydrogen with oxygen from the air to produce steam or water. The output depends upon the type of fuel cell, electrolyte and the electrical current used.

The advantages of the fuel cells are high efficiencies, low load conditions, no wear and tear of the components, no or low emissions (since pure hydrogen or natural gas is used). Fuel cells are quiet and transportable. The available sizes are in the range of 5kW-2MW [6].

1.6.1.7 Stirling engines based cogeneration systems

Stirling engines are external combustion engines. Fuel is burned outside of the engine cylinder which has a working fluid which burns continuously. Since the fuel is burnt outside of the engine cylinder it has flexibility in the type of the fuel to be used. It also has good heat recovery.

Table.1.1. Comparison of Cogeneration systems [Source: UNESCAP and the European Association for the Promotion of Cogeneration [6].

Cogeneration System	**Electrical energy output** (% of fuel input)	**Overall efficiency** (%)	**Heat to power ratio**	**Thermal Qualities**
Back pressure steam turbine	14-28	84-92	4.0-22.0	High
Condensing steam turbine	22-40	60-80	2.0-10.0	High
Gas turbines	24-42	70-85	1.3-2.0	High
Resiprocating engine	33-53	75-85	0.5-2.5	Low
Combined cycle gas turbine	34-55	69-83	1.0-1.7	Medium
Fuel cells	40-70	75-85	0.33-1	low to high
Microturbines	15-33	60-75	1.3-2.0	Medium to low
Sterling engines	05.-45	over 80	1.76-2.4	Medium to low

The thermal efficiency of the Stirling engine is same as the conventional steam boiler. The working fluid in the Stirling engine never leaves the engine so engine becomes less complicated (no exhaust valves). The engines run quietly. These engines have high heat to power ratio which makes them suitable for load requirements in residential

applications. The following Table.1.1 summarizes the performance characteristics of various cogeneration systems.

1.7 Gas Turbine Cogeneration Systems

Gas turbine cogeneration systems are more popular among all other power generation systems, since these systems have inherent advantages such as high efficiency, clean, flexibility in terms of the type of fuel used, low production and maintenance cost as well as lower greenhouse gas emissions.

In gas turbine cogeneration systems, gas turbine is main component which drives the system. Gas turbines are well established/ proven prime movers. The exhaust heat of a gas turbine can be used for district heating and cooling applications. This high temperature heat can be recovered to improve system efficiency or can be utilized to generate high quality steam. This steam is to power steam turbine in a combined cycle plant. The emissions from gas turbine exhaust can be reduced to low level by dry combustion technique, water or steam injection or exhaust treatment. Maintenance cost per unit power of the gas turbines is less than that of internal combustion engines [7]. Considering maintenance cost and high quality exhaust, gas turbine systems are the better choice for power output greater than 3 MW.

As a prime mover, the gas turbines are increasingly used in one of the following ways.

- In a cycle where only power is produced by a gas turbine.
- In a CHP configuration with HRSG, in which the heat from the exhaust gas is recovered to produce steam or hot water.
- In a combined cycle, in which high steam produced from the gas turbine exhaust gases is increased to generate additional power using steam turbines. In some applications, the steam is extracted at the increased pressure to cater for process or industrial applications. These are combined cycles with CHP application. The gas and

steam turbine, combined cycle is the most efficient system for power generation, which gives higher efficiencies.

The efficiency of a basic cycle with a gas turbine for power only applications is about 40 %. In a combined heat and power configuration, due to utilization of the high quality gas turbine exhaust, the combined heat and power efficiencies are up to 70 -80 % [7].

In the past, gas turbine systems were used for peak load conditions but due to technological developments they have been used for base load conditions as well.
Gas turbine systems are based on the Brayton cycle [5]. In the Brayton cycle, the air at ambient temperature is compressed in a compressor, heated in a combustor and then expanded in a turbine to generate power. The power generated is proportional to the absolute temperature of the working media in the components. In gas turbines in order to get maximum efficiency, the turbines are run on maximum practical inlet temperatures. The practical constraints in achieving maximum possible temperatures are the selection of turbine blade material that can withstand the high temperatures. On the other hand, in the compressors inlet temperatures of the air should be as low as possible.

Gas turbines are preferred for CHP applications due to their high temperature exhaust which can be increased to generate thermal energy at 1200 psig and 900° F [7].

1.8 Advantages of Gas Turbine Cogeneration

These systems also offer environmental benefits. They typically produce very low emissions, virtually no carbon monoxide or sulfur dioxide emissions, and since natural gas derives significant energy from burning hydrogen, much less carbon dioxide than coal. All these factors make gas turbines suitable for urban areas, where the plant operator can achieve an extra savings in fuel and emissions by avoiding transmission losses. More benefits come from fuel efficiency. A base-load coal plant may convert up

to 35 percent of available energy to electricity [3-9]. Gas turbines already operate at basic-cycle efficiencies of above 40 percent; combined-cycle ratios approach 60 percent less fuel is used [3-9], and the fuel used burns cleaner. Efficient heat rates can help overcome higher energy costs.

The natural gas based power plants have competitive edge over coal based power plants with respect to efficiency achievements topping 60 % [7].

a) Lower capital costs $600-$750 per KW compared with $1400-$2000 per KW for typical coal based power plant [6].
b) Shorter construction times.
c) Environmental friendly. The environmental benefits such as:

- Lower emissions: natural gas based power plant emits fifty percent less than similar rated coal based power plant.
- Reduced sludge**:** Natural gas based power plants emit drastically low levels of SO_2, thereby eliminating need of scrubbers and reducing amount of sludge formation.
- Reburning: In reburning process the natural gas is injected into coal or oil fired boilers. Due to injection of natural gas the NO_X emission level drops and lowers the SO_2 emissions.
- Cogeneration: Natural gas is the preferred gas for cogeneration. The combined effect of the production and use of heat and electricity delivers benefits on fuel consumption and emissions.
- Combined cycle generation: The thermal efficiency of these plants is much higher than conventional coal or oil. This is due to the fact that combined cycle utilizes waste heat energy to generate more electricity.
- Fuel cell**:** Natural gas based fuel cells are still in development stage. Natural gas being a rich source of hydrogen can be used as fuel.

1.9 Methods to Improve the Performance of Cogeneration Systems

1.9.1 Intercooling

In this process, air is compressed in stages and cooled between stages. Thus the work required for compression reduces due to less temperature difference. Since a compressor takes less energy, the net power output increases. As a result the efficiency of the cogeneration system increases.

Intercooling may look promising but it has a great disadvantage since the combustor needs to do more work. It also needs to compensate the heat that is taken away by intercooling. Thus fuel consumption increases. Intercooling therefore increases net work output at the cost of efficiency [5 -8].

1.9.2 Reheating

The idea of reheating is same as intercooling but applies to the turbine. In this process, the expansion of the working media in turbine is done in stages. The working media is reheated at every stage. Due to the reheating, the work output and efficiency increases. Reheating has similar disadvantages as intercooling. It increases the work output by reducing efficiency [5-8].

1.9.3 Regeneration

In this process a heat exchanger is used. The heat exchanger is also called recuperator. The heat of the exhaust turbine gas is utilized to warm up the air /gas entering the combustor. The regeneration process can only be applied in the situations when

a. Turbine exhaust temperature is more than air temperature at the compressor outlet and,
b. The pressure ratio is less than optimum pressure ratio.

That means that the heat exchangers are beneficial for low pressure ratios.

Thus the efficiency increases, as the heat input is reduced for same output.

Thermodynamic efficiency [5] is given by

$$\eta_{Th} = 1 - \left(\frac{T_1}{T_3} r_p^{\frac{k-1}{k}} \right) \quad (1.1)$$

1.9.4 Raising turbine inlet turbine

The gas turbine efficiency depends on the ratio of gas turbine inlet temperature to compressor inlet temperature. Raising TIT is definitely an effective way to increase the power and efficiency of cogeneration system; however it is usually limited by material strength to resist heat / temperature and better cooling technology for the components of the turbine [5].

1.9.5 Enhancing compressor and turbine efficiencies

Increased efficiencies of the compressor and turbine in turn increase the efficiency of the cycle. CHP and combined cycles are two most common practices in recovering heat energy from the exhaust of the gas turbine [8].

1.9.6 Steam injection

The steam generated from CHP, can also be used to improve the power efficiency and capacity. The generated steam can be directly injected into the combustor to increase turbine efficiency significantly. These are steam injected gas turbine (STIG). The increased mass flow rate due to injection causes to increase the power generation efficiency. This phenomenon was first realized by Cheng [35], who discovered that by steam injection the power output, can be increased by 75% and thermal efficiency by 40% [35].The steam injection is now a well-established practice [21].

The advantages of steam injected gas turbine systems are such as low capital cost, operational flexibility as per power and heat demand, and reliability makes steam injection gas turbine cogeneration system an attractive electrical generating technology.

1.10 Thesis Objectives

In this thesis work, performance of two gas turbine cogeneration systems, the basic gas turbine cogeneration system and steam injected gas turbine cogeneration systems, are analyzed based on energy and exergy analysis. The main objectives of this thesis are:

- To perform parametric analysis of GT cogeneration systems to study the effect of gas turbine inlet temperature, pressure ratio and efficiencies such as power, heat generation and cogeneration.
- To investigate the effect of steam injection on specific power, energy and exergy efficiencies and CO_2 emissions.
- To assess the impact of specific heat variation on performance parameters of a gas turbine cogeneration system.
- To conduct exergy analysis of gas turbine cogeneration system from second law point of view.
-

1.11 Outline of the Thesis

This thesis is organized into five chapters. Chapter 2 reviews the literature pertaining to gas turbine cogeneration systems. A synopsis of the relevant and current literature is presented.

Chapter 3 includes methodology and system description in which thermodynamic principles including the concept of exergy is discussed. In the gas turbine system description, system diagrams, data and assumptions are provided. In the analysis, each component of the system is modeled and analyzed. The mass balance, energy and entropy balance equations are developed. The exergy analysis includes formulation of exergy balance equations, irreversibility, power, heat and total exergy efficiencies.

In Chapter 4, the results are discussed and the performance characteristics of both systems are evaluated by varying the operating variables such as pressure ratios and turbine inlet temperatures. The characteristics of the BGCS and the STIG are compared with reference to the heat demand ratio which is the ratio of heat demand to the maximum heat supplied. Further, the impact of steam injection on emissions is determined. The chapter concludes by examining the role of the specific heat in the analysis.

Chapter 5 contains the concluding remarks, the contributions to this study and recommendations for future work.

CHAPTER 2. LITERATURE REVIEW

Cogeneration is a very old concept. References can be found from the beginning of the twentieth century. In fact, the first commercial power plant was built in Manhattan in 1882, which is Thomas Edison pearl street station [10]. At the time reciprocating steam engine was the dominating prime mover used for generating electricity and low pressure exhaust steam was used for heating applications. During 1970 -1985 the importance of cogeneration grew significantly due to the energy crisis in America and price of the fuel and power that escalated by a factor of five. Cogeneration is a distributed generation system. There has been growing interest in distributed generation in North America, since US and Canada experienced a major power outage in August 2003 that lasted for several days.

Various combination of gas turbine cogeneration systems are studied and analyzed by many researchers. The factors affecting gas turbine performance are reviewed by Brooks [11]. The performance results of several models of General Electric (GE) products are discussed in that study. The characteristics of basic gas turbines are discussed. The study is conducted on one of the GE's product .The effect of steam and water injection on power augmentation, are investigated and summarized.

Exergy analysis of gas turbine cogeneration systems was reported in the simulation studies by Pak and Suzuki [12]. It was pointed out that the exergetic efficiency increasesd with increase in heat ratio in dual fluid cycle. Power generation is directly proportional to the injected steam whereas inversely proportional to the heat generation. The data generated by comparison between duel fluid cycle and combined cycle is helpful in selection of appropriate cogeneration.

A natural gas turbine cogeneration system is analyzed by Reddy and Cliff Butcher [13]. They reported the effects of operating conditions such as reheat, intercooling, ambient temperature and pressure ratio. They also reported that the second law efficiency of the

system decreases with higher pressure ratios, which can be compensated by utilizing the reheat and inter cooling options to improve the cogeneration efficiency. The importance of second law analysis in achieving energy productivity is discussed by Bilgen et al. [14]. They analyzed a gas turbine cogeneration system with the help of thermodynamic principles of conservation of mass and energy along with the second law. In the second law analysis calculation methodology to determine physical and chemical exergy were described. Heat recovery steam generator design parameters were also investigated with reference to process steam pressure and pinch point temperature by Khaliq and Kaushik [15]. It was pointed out that the power to heat ratio of gas turbine cogeneration system is proportional to pinch point temperature whereas the exergy and energy efficiencies are inversely proportional. Their study showed how reheat improves electrical output, process heat, and energy and exergy efficiencies. The effect of compressor pressure ratio and turbine inlet temperature on exergy destruction was also examined. It was also concluded that performance evaluation based only on first law is not adequate and second law analysis is necessary to get meaningful inferences.

In order to increase the power output of a gas turbine cogeneration system**,** Khaliq and Choudhary [16] proposed inlet air cooling. It was shown that effectiveness of heat exchanger can be raised by evaporative after cooling of the compressor discharge. It inferred that the maximum exergy destruction takes place in the combustion chamber is 70 % of total exergy destruction. The percentage of exergy destruction at the compressor and turbine is relatively low and has less influence on the cogeneration system.

Analysis of entropy generation in HRSG, considering non-dimensional operating parameters is studied in detail by Reddy et al. [17]. Considering all irreversibility in the process, an entropy equation was proposed. The influence of parameters such as specific heat of gas, inlet gas temperature ratio and heat exchanger unit sizes on entropy was examined to obtain the optimum values.

In the energy analysis of gas turbine cogeneration systems, Rahman et al. [18] concluded that the thermal efficiency of a gas turbine cogeneration system is greatly affected by the operating variables such as pressure ratio, ambient temperature, air fuel ratio and isentropic efficiencies of the components.

Ganapati [19] confirmed that the exit gas temperature of the gas turbine influences the HRSG efficiency. He recommended several options to improve HRSG efficiency. He discussed the temperature profile diagram and developed relationships among gas temperature at several stages in HRSG and pinch point.

Parametric studies of regenerative cogeneration plant are conducted by Ashok Kumar et al. [20]. In that study thermodynamic performance was evaluated based on first law efficiency, second law efficiency, heat to power ratio and specific fuel consumption by changing variables such as pressure ratio, turbine inlet temperature, steam pressure and pinch point temperature. It is also indicated that the optimum pressure ratio found to be 20 at which first law and second law efficiencies were at maximum, and specific fuel consumption was minimum. The cogeneration system efficiency and power to heat ratio are increased with turbine inlet temperature. The exergy analysis revealed that the combustion process was sensitive to the exergy destruction and had the maximum destruction (60-85%).

In the analysis of steam injected gas turbine combined cycle Srinivas et al. [21] discussed the calculation methodology in detail. The effect of steam injection on the performance of the combined cycle was examined. In their calculations the steam injection ratio was considered with respect to the mass of fuel. It was indicated that the gas cycle efficiencies and combined cycle improves with increase in steam injection ratio.

The steam injection with reheat and intercooling options were investigated on aero gas turbine engine by Rice [22]. In his assessment, he discovered that the ratio of

compressor work to the turbine output decreases with increase in turbine inlet temperature.

The performance characteristics of the steam injected gas turbine cogeneration system were investigated by Mohammed et al. [23]. The energy and exergy analysis showed that the power output increased with steam injection at the same time cogeneration efficiency decreased with increasing amount of steam injection.

Several humidified gas turbine cycles are reviewed by Jonson et al. [24]. It was indicated that the specific work output is augmented due to increase in mass flow rate in the turbine. As the compressor work remains constant, the cycle efficiency can be increased by utilizing heat energy of the exhaust of the turbine to generate steam for injection or preheating the compressed air prior to combustion (recuperation). It is also evolved that the steam or water injection reduces NOx emissions.

Aissani et al. [25] evaluated the performance characteristics of steam injected gas turbine cogeneration system**.** In their study carbon monoxide and nitrogen oxide emissions were assessed in a steam injected gas turbine cogeneration system. It was proved that the steam injection reduced NO_x formation which helped clean the environment.

Optimal operating conditions for maximum thermal efficiency for regenerative steam injection cogeneration system and steam injection system were assessed by Kim et al. [26]. The performance parameters such as thermal efficiency, specific fuel consumption specific power were evaluated by varying the operating variables. In a steam injection system, the fuel consumption decreases with increase in the pressure ratio, since compressor outlet temperature increases with increase in pressure ratio. For fixed turbine inlet temperature, the required fuel quantity decreases and thus fuel consumption.

The retrofitted steam injected gas turbine cogeneration system was analyzed by Wang et al. [27]. Their study showed that steam injection effectively boosted both power output and thermal efficiency compared with basic gas turbine cogeneration system. Manshoori et al. [28] investigated a steam injection system in a gas turbine system. Their study revealed that, the steam injection helped reduce the GHG emissions of NOx and CO_2 although CO increased.

Motahar et al. [29] conducted energy and exergy analysis of steam injected gas turbines with evaporative cooling system. Their analysis indicated that from thermodynamic point of view STIG systems were more efficient and suitable for CHP applications.

Agarwal and Mishra [30] examined various retrofitting techniques such as inlet air evaporative cooling, STIG, steam injection combined with inlet air cooling, to improve performance of a simple gas turbine cycle. In the exergy analysis they concluded that STIG option was better than among other techniques. The STIG technology enhances the power output, thermal efficiency and fuel air ratio.

Several steam injection configurations including, water and steam injection regenerative steam injection gas turbines are studied by Nishida et al. [31]. It is indicated that these configurations can be applied to small scale gas turbines. The characteristics of the systems revealed that the steam injection systems can be used for flexible heat and electrical power demand.

Cogenerations help reduce the fuel consumption in power plant, thereby decrease in GHGs. This beneficial aspect was reported by Rosen [32] by considering a hypothetical case of electricity utility cogeneration. In this study, it was observed that the fuel consumption reduced by 10 – 40 % coal; 3 to 35% uranium; and 10 to 40% carbon dioxide emissions. Since cogeneration offers significant reduction in fuel usage therefore

improvement in fuel utilization, energy efficiency and less emissions that means better environment.

Dincer [33] presented the relationship among the environmental impact, thermodynamics and exergy. The concept of exergy was elaborated. It was concluded that, the second law analysis of thermal systems was critical in addressing and solving environmental issues.

Dincer [34] discussed the environmental impact of global energy production and consumption as well as the symptoms of global warming such as climate change, acid rain and air pollution. It was commented that the main drivers of increase in energy consumption including population growth and economic growth. Furthermore energy conservation and various fuel options were also discussed.

Tara Chand et al. [37] studied the exergy analysis of gas turbine power plants. In the analysis it is shown that the combustion chamber has the highest irreversibilities among rest of the components of gas turbine power plant; however it is also shown that total irreversibilities are reduced due to higher pressure ratios and turbine inlet temperatures.

Atouei et al. [38] analyzed steam injected gas turbine systems with energy, exergy and exergoeconomic point of view. In their study it is shown that steam injected gas turbine system is more efficient than simple gas turbine systems. In exergoeconomic analysis it is shown that due to optimization the exergy destruction cost decreased but at the same time capita cost per hour increased.

Although previous investigations provided some insight into energy and exergy analysis of gas turbine cogeneration systems, few studies were conducted on the steam injection

in gas turbine cogeneration systems and its impact on CO_2 emissions. There is need for the performance analysis of gas turbine cogeneration systems for better understanding and improvements. The driving force for further study of the steam/water injected gas turbine (humidification) has the potential of high electrical efficiency and specific power output as well as reduced specific investment cost, decreased formation of emissions in combustor and improved part load performance compared with basic cogeneration system.

CHAPTER 3. GAS TURBINE SYSTEM DESCRIPTION AND METHODOLOGY

3.1 Introduction

One of the common the gas turbine cogeneration systems used is a basic type that operates on Brayton cycle [5]. These systems are suitable for wide industrial applications. In order to further improve the performance of gas turbine cogeneration system and establish the basis for the comparison the basic type of gas turbine cogeneration system is chosen for analysis in the first part of the wok.

In the literature various combinations of gas turbine cogeneration systems have been studied and proposed in order to improve the performance of gas turbine cogeneration systems. The steam injection in the combustion chamber is a well-established practice, especially in a small medium range size operating under varying heat and power demand. Earlier the steam injected gas turbine cogeneration systems are analyzed with regards to power augmentation, reductions in NOx. In the light of current climate change scenario, the CO_2 emission is the biggest concern. In order to study the performance improvement due to the steam injection and influence of operating variable on the CO_2 emissions, this configuration of steam injected gas turbine cogeneration system is chosen.

In this study, two gas turbine cogeneration systems are described. The first one is a basic gas turbine cogeneration system with HRSG. In it all the steam generated in the HRSG is utilized for process application. The second system is a steam injected gas turbine cogeneration. In this system, part of the steam generated in the HRSG is injected in the combustion chamber to increase the turbine output. The rest of the steam can be utilized for process application. In this investigation, systems performance assessment is done based on first law and second law analysis. The effect of steam injection on the performance characteristics is also examined.

The heat losses through equipment are categorized as exergy destruction. The exergy destruction as the name suggests is the opposite of entropy generation. The stack temperature is controlled due to corrosion considerations. Typically the stack temperature is taken as 120℃ [5-8].The methodology used to evaluate the cogeneration systems is well established. The mathematical modeling of each component is presented. Mass, energy and entropy balance equations are established for each component in the system compressor, combustion chamber, turbine and HRSG. Irreversibility and exergy destructions of each component are also calculated.

3.2 System Description and Methodology

3.2.1 Basic gas turbine cogeneration system (BGCS)

The sketch shows typical gas turbine based cogeneration plant. It consists of a compressor in which air with ambient parameters (P, T) is compressed and fed to a combustion chamber. The optimal pressure ratio is chosen with regards to the performance of the cogeneration system. The compressed air with elevated pressure and temperature fed into combustion chamber along with fuel combustion takes place. The product of combustion is then fed to the turbine. The hot exhaust gas is used as media to run the turbine. Turbine produces power which is used to run the compressor and remaining power is utilized in power applications. The expanded hot gas from the turbine is then fed through heat recovery steam generator. The heat energy from the turbine exhaust is further extracted in HRSG. The extracted heat energy is utilized to generate saturated steam. The steam is then used for heating and cooling applications.

The gas turbine cogeneration system operates on Brayton cycle. Referring to the schematic in Figure 3.1, Path 1-2 represents the compression in the compressor, Path 2-3 represents the combustion in combustion chamber which is constant pressure heat addition; Path 3-4 represents expansion of gases in turbine which gives the work output and Path 6-7 represents the steam generation in HRSG.

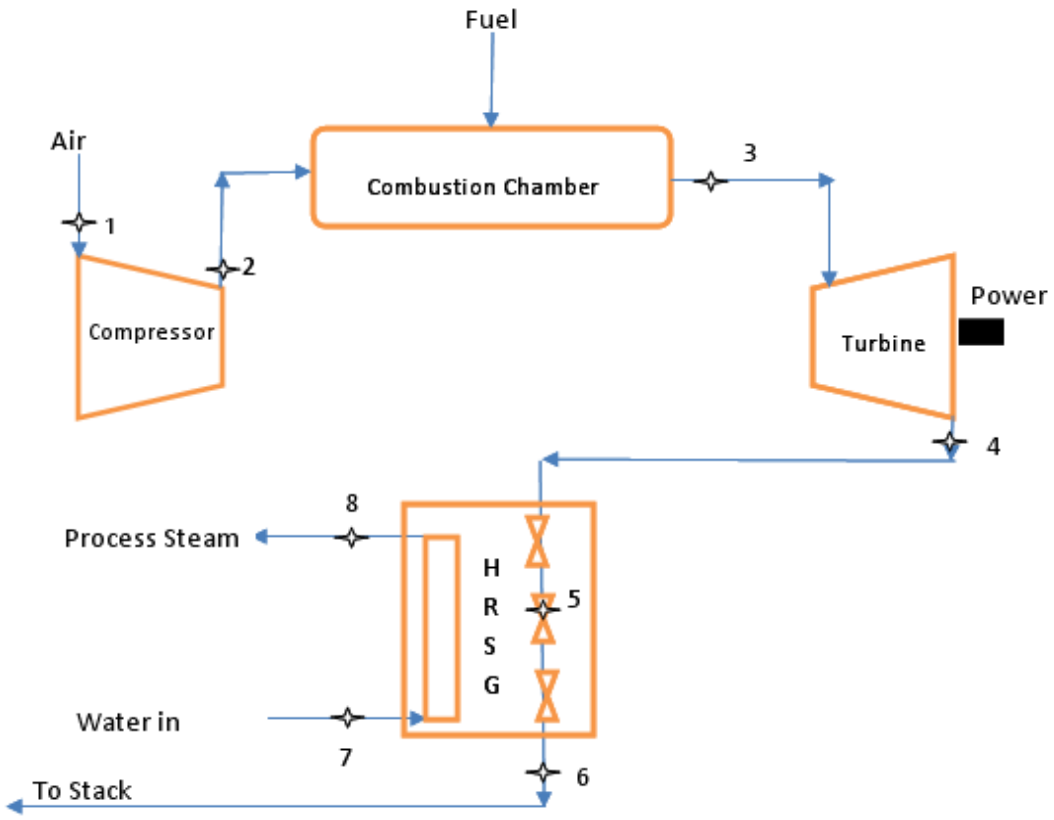


Figure 3.1 Schematic diagram of the basic gas turbine cogeneration system

Practically, gas turbine systems are open cycle as opposed to the closed cycle as shown in Figure 3.1. The exhaust gases from HRSG are sent into the atmosphere but at the Stage 1, fresh cool air is sucked by the compressor [36].

In general, the performance of gas turbine cogeneration system depends upon the process heat demand. The BGCS is efficient for high heat demand; the performance deteriorates for less process heat demand. Since less process heat demand means less fuel requirement which in turn affects power output. Thus low process heat demand decreases net power output significantly.

3.2.2 Steam injected gas turbine cogeneration system (STIG)

The schematic in Figure 3.2 shows steam injected cogeneration system. The primary function of steam injection is to raise the specific power output [11-36]. The concept of

steam injection is based on the fact that the injected steam increases the mass flow rate through the turbine which increases the power output. In this system, the saturated steam which is generated in HRSG is injected in the combustion chamber. The enthalpy of the super-heated steam is utilized in the combustion products. In the STIG cycle the compressor work remains constant as the steam is injected after the compression process. The STIG cycle efficiency increases when the heat energy from turbine exhaust gases is utilized for steam injection and heating /cooling applications.

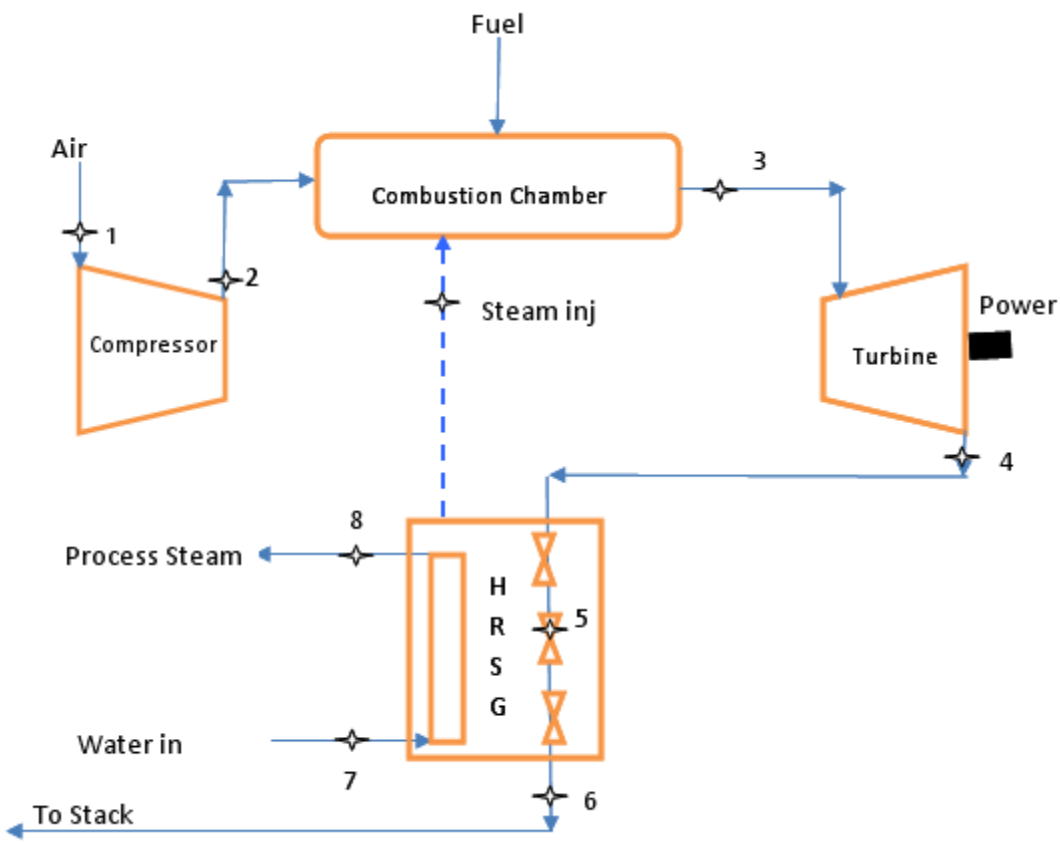


Figure 3.2 Schematic diagram of the steam injected gas turbine cogeneration system.

The construction of STIG is similar to the BCGS except a provision of steam injection from HRSG to the combustion chamber. The steam injection pressure should be equal or more than compressor outlet pressure [31]. This system is not sensitive to the lower heat demand because at lower process heat demand the excess steam can be injected

back in the combustor to augment the power. In this type of cogeneration, the enthalpy of the steam is utilized to increase the power output. Conversely, for the fixed power output and fixed pressure ratio the fuel consumption drops thereby decreasing emissions.

Since the flow rate of the steam is too low compared with air flow rate, for the calculation purpose the change in pressure due to steam injection is neglected. The performance of both of these systems is analyzed thermodynamically and performance characteristics are established.

3.3 Assumptions

Appropriate assumptions were made in order to simplify and standardize the problem, and to estimate performance characteristics of the cogeneration systems with regards to ambient state, adiabatic and isentropic component efficiencies and; ideal gas and combustion.

In this analysis, the following assumptions were made in order to apply the thermodynamic laws and concepts to the case studies

i. Complete combustion in combustion chamber.
ii. There is no heat loss in compressor, combustor, turbine, HRSG and its accessories.
iii. Ambient temperature and pressure is considered as 0.101 MPa and 25 ℃
iv. Isentropic efficiencies of turbine and compressor are taken as 85%.
v. The chemical exergy of methane is considered in exergy analysis.

In this analysis the combustion with lean fuel that is with excess air is assumed. The complete combustion generates carbon dioxide whereas fuel rich combustion results in carbon monoxide in the products of combustion. Since one of the objectives of this study is to evaluate carbon dioxide emissions, the complete combustion is assumed.

The heat losses due to leakages are generally assumed as two percent of the total losses. The value of heat loss due to leakages is significantly less as compared to the total losses and irreversibilities. So the heat losses due to leakages are neglected.

The isentropic efficiencies are the thermodynamic efficiencies which compares real behavior to the ideal behavior of the working fluid. In the ideal behavior is related to the absence of thermodynamic losses, referred to as irreversibilities. In real process however irreversibilities are present due to friction and other internal losses (second law of thermodynamics). In compressor actual work consumed is more than ideal work required in compression, whereas in the gas turbine the generated output is less than the isentropic work due to the ireversibilities. In order to represent the real compression and expansion processes the isentropic efficiencies are assumed. Typically the isentropic efficiencies are assumed as 0.85 [5].

In exergy analysis standard values of chemical exergy values of chemical elements at standard ambient temperatures are taken from thermodynamic book [5].

3.4 Basic Gas Turbine Cogeneration System Analysis and Methodolgy

In the analysis, the components of basic gas turbine cogeneration system, compressor, combustion chamber, turbine, and heat recovery steam generator. Thermodynamic analysis is performed based on energy and exergy analyses. In the energy analysis, mass, energy and entropy balance equations are developed. The irreversibility in the BGCS is determined with the help of exergy analysis.

3.4.1 Air Compressor in BGCS

To analyze the compressor of the basic gas turbine cogeneration system a pressure ratio is selected. Based on the pressure ratio, the outlet temperature is evaluated. The

ambient temperature and pressure are assumed. In order to evaluate the outlet temperature of the compressor, it is assumed that the air is compressed isentropically.

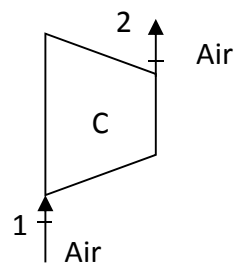


Figure 3.3 Schematic diagram of compressor of the basic GT cogeneration system

The temperature T_{2S} is thus estimated.

$$\frac{T_{2s}}{T_1} = (r_p)^{\frac{k-1}{k}}, \tag{3.1}$$

where r_p – pressure ratio, and k- specific heat ratio

In an alternate method, the entropy change in the isentropic compression process is zero.

$$\Delta s = s_{2s} - s_1 = 0 \tag{3.2}$$

$$s_{2s,O_2} - s_{1,O_2} + 3.76(s_{2s,N_2} - s_{1,N_2}) - R[\ln(0.21r_p) + 3.76\ln(0.79r_p)] = 0, \tag{3.3}$$

where, R - Universal Gas Constant

The isentropic outlet temperature at the compressor (T_{2s}) is estimated by iteration method. The actual temperature can be calculated by isentropic efficiency and enthalpy. The isentropic efficiency of compressor is considered to be within the range of 85%.The actual temperature T_2 is determined by Equation [3.4].

$$\eta_{compressor} = \left(\frac{T_{2s} - T_1}{T_2 - T_1} \right) \tag{3.4}$$

In energy analysis of compressor mass, energy and entropy balance equations are developed.

Referring to the Figure 3.3, the mass of ambient air entering into the compressor and mass of compressed air going into the cogeneration system are considered equal. It is assumed that there are no leakages at the compression stage. Thus mass balance equation at the"1 "and "2"states can be written as:

$$\dot{m}_2 = \dot{m}_1 = \dot{m}_a \tag{3.5}$$

This is mass of air in kg/s.

Similarly, considering the enthalpy of air at the inlet of the compressor and enthalpy of compressed air, the energy balance equation can be developed to obtain work output required for compressor. It is assumed that the compressor is adiabatic. Thus energy balance equation at the "1" and "2"states can be formulated as

$$\dot{m}_1 * h_1 = \dot{m}_2 * h_2 - W_c , \tag{3.6}$$

where, W_C - compressor work. Alternatively the equation can be written as:

$$W_c = \dot{m}_a c_{pa} (T_2 - T_1) \tag{3.7}$$

Similarly, by equating entropy at the "1"and "2" states; the change in entropy ($s_{gen.c}$) during compression can be found.

$$s_{gen.c} = \dot{m}_a (s_2 - s_1) \tag{3.8}$$

$$s_2 - s_1 = c_{pa} \ln\left(\frac{T_2}{T_1}\right) - \frac{k-1}{k}\left(\ln\left(\frac{P_2}{P_1}\right)\right), \tag{3.9}$$

where s_1 and s_2 are the specific entropies at given state.

Irreversibility in compressor (i_{comp}) can thus be evaluated.

$$i_{comp} = T_0 . s_{genc} \tag{3.10}$$

The exergy at the compressor stage can be found out by applying exergy balance equation at “1 and “2” states. The exergy in at the state “1” is equal to the exergy out at “2”. This can be formulated as:

$$\dot{e}_{x1} + i_{comp} = \dot{e}_{x2} + W_C \; ; \tag{3.11}$$

Where,

$\dot{e}_{x1}$ and $\dot{e}_{x2}$ are specific physical exergy at the given state 1&2;

W_C is the work output consumed by the compressor;

i_{comp} is the irreversibility at the compression stage due to elevated temperature.

3.4.2 Combustion chamber in BGCS

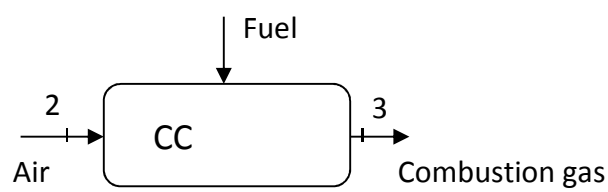


Figure 3.4 Schematic diagram of combustion chamber of basic GT cogeneration system

The pressurized hot air from the compressor enters the combustion chamber where fuel is injected. The combustion product which is the working media is expanded through the turbine to produce power. It is assumed that the combustion takes place adiabatically with constant pressure. In the analysis, the turbine inlet temperature T_3 is kept constant (1273 K). The fuel used is methane gas. The products of the combustion are carbon dioxide, water vapor, nitrogen and heat energy. Practically there are no specific or definite ideal combustion conditions. Specific amount of oxygen is needed to burn the fuel and that is the theoretical air required to complete the combustion.

Insufficient air in combustion chamber leads to unburnt fuel, soot smoke, carbon monoxide which affects heat transfer, pollution, the system performance and efficiency. In this study, the specific heats of combustion products are determined by adding specific heat of each compound in the products of combustion in combustion gas mixture temperature and the composition of gases in terms of mole or mass fraction. The average specific heat for the combustion gas has been determined from the variable specific heat equations.

For excess air calculations, one must know the Stoichiometric air fuel ratio. The Stoichiometric air fuel ratio is the air fuel ratio with chemically correct proportion. During combustion process the whole air and fuel is consumed. In practice the excess air is always fed to the combustion chamber to burn the fuel completely. To avoid this, the excess air is provided.

The Stoichiometric equation for the combustion of methane is as follows [5-8]:

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2 \qquad (3.12)$$

In practice, more than theoretical air is supplied to burn the fuel completely. Gas turbines run very lean up to 300 percent excess air. The Stoichiometric air fuel ratio for methane is 17.241. In energy analysis of combustion chamber, mass, energy and entropy balance equations are developed.

Referring to Figure 3.4, the mass entering the combustor and the mass going out of the combustion chamber can be balanced. In the combustion chamber the compressed air and the fuel get mixed in the combustion reaction. The hot combustion gas formed as a result of combustion reactions goes out of the combustion chamber. It is assumed that the combustion chamber is leak proof. Thus mass balance equation at the combustion stage “2”and “3”can be formulated as:

$$\dot{m}_a + \dot{m}_f = \dot{m}_g, \qquad (3.13)$$

The mass balance equation is formulated based on the assumption that 1 Kg of combustion gas is formed during combustion. This product of combustion would be the working medium in the cycle.

In combustion process the enthalpy of the compressed air and enthalpy of injected fuel (calorific value) play important role in the chemical reaction which produces the hot combustion gas with very high enthalpy. Thus the energy balance at state"2"and "3"can be formulated as:

$$\dot{m}_2 h_2 + \dot{m}_f h_f = \dot{m}_g h_3, \tag{3.14}$$

where, h_f is the enthalpy of injected fuel which is lower calorific value of the fuel. The fuel consumption for 1 Kg/s of combustion gas the equation can be simplified in terms of specific heat as:

$$\dot{m}_f = \frac{\left(c_{pg3} T_3 - c_{pa} T_2\right)}{\left(CV - c_{pa} T_2\right)} \tag{3.15}$$

where $\dot{m}_f$ is rate of fuel consumption (kg/s).

Using Equation 3.13, m_a can be calculated and the air fuel ratio is determined.

Alternatively, the amount of air in the combustion reaction can be found out by balancing combustion equation:

Let us consider 1kmol of methane gas is used in combustion process. Also assume, adiabatic combustion takes place in combustion chamber. The energy balance in the combustion chamber is:

$$CH_4 + X(O_2 + 3.76N_2) = b_1CO_2 + b_2H_2O + b_3O_2 + b_4N_2 \tag{3.16}$$

The amount of air "X" in moles for fixed adiabatic flame temperature T_3 is determined as follows. The coefficients of the chemical equation are calculated by balancing the Equation 3.16.

$b_1 = 1$; $b_2 = 2$; $b_3 = X - 2$; $b_4 = 3.76X$. (3.17)

The energy balance across the combustion chamber is expressed as follows:

Sum of enthalpies of reactants = Sum of enthalpies of the products or

$H_R = H_P + \cancel{Q_{out}}$ (adiabatic) (3.18)

$$\left(h_{f,CH_4} + h_{CH_4}\right)_{T_2} + X\left(h_{O_2} + 3.76h_{N_2}\right)_{T_2} = \left(h_{f,CO_2} + h_{CO_2}\right)_{T_3} + 2(h_{f,H_2O} + h_{H_2O})_{T_3} +$$

$$(X-2)h_{O_2,T_3} + 3.76xh_{N_2,T_3} \tag{3.19}$$

Equation 3.19 can further be simplified to

$$X = \frac{\left(\left(h_{f,CH_4} + h_{CH_4}\right)_{T_2} - \left(h_{f,CO_2} + h_{CO_2}\right)_{T_3} - 2\left(h_{f,H_2O} + h_{H_2P}\right)_{T_3} + 2h_{O_2,T_3}\right)}{h_{O_2,T_3} + 3.76h_{N_2,T_3} - \left(h_{O_2} + 3.76h_{N_2}\right)_{T_2}} \tag{3.20}$$

In the exergy analysis of combustion chamber, in order to determine exergy of reactants and products, both physical and chemical exergy are considered.

$$e = e_{ph} + e_{chem} \tag{3.21}$$

In terms of physical exergy and chemical exergy Equation 3.21 can be written as

$$\left(e_{ph} + e_{chem}\right)_{air} + \left(e_{ph} + e_{chem}\right)_{fuel} = \left(e_{ph} + e_{chem}\right)_{comb.gas} \tag{3.22}$$

In physical exergy calculations both components of pressure and temperature are considered. The physical exergy of air determined by [5]:

$$e_{phair} = X\left(\sum_k n_k(h_T - h_{298}) - T_0\left(\sum_k n_k(s_T - s_{298}) - n_{air}R\ln\left(\frac{p}{p_0}\right)\right)\right) \tag{3.23}$$

The chemical exergy is expressed as [6]:

$$e_{chair} = X * n_{air}\left(\sum_k X_k\varepsilon_k^0 + RT_0\sum_K X_k * \ln(X_k)\right) \tag{3.24}$$

At ambient temperature the physical exergy of methane is considered as zero [6]. The chemical exergy of the methane is the standard value, which is 836510 kJ / kmol.

The physical exergy of products of combustion is determined as [5]:

$$e_{ph_{products}} = \left[\sum_k n_k \left(h_{T_3} - h_{298} \right) - T_0 \left(\sum_k n_k (s_{T_3} - s_{298}) - n_p R \ln\left(\frac{p}{p_0} \right) \right) \right] \tag{3.25}$$

The chemical exergy of product of combustion is determined by [5]:

$$e_{ch_{products}} = n_p \left[x_{CO_2} \varepsilon^0_{co_2} + x_{H_2O} \varepsilon^0_{H_2O} + x_{O_2} \varepsilon^0_{O_2} + x_{N_2} \varepsilon^0_{N_2} \right] + n_p \left[RT_0 \left(x_{CO_2} \right) + x_{H_2O} \ln\left(x_{H_2O} \right) + x_{O_2} \ln\left(x_{O_2} \right) + x_{N_2} \ln\left(x_{N_2} \right) \right] \tag{3.26}$$

The irreversibility / exergy destruction in the combustion chamber is then determined by balancing exergy in and exergy out. Referring to the figure 3.4, the exergy balance in combustion chamber is expressed as:

$$\dot{e}_{air} + \dot{e}_{fuel} = \dot{e}_{combgas} + i_{cc} \tag{3.27}$$

3.4.3 Gas turbine in BGCS

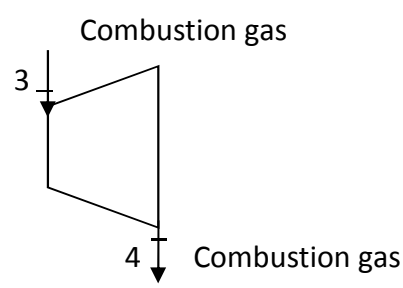


Figure 3.5 Schematic diagram of gas turbine of the basic GT cogeneration

Gas turbine generates power by expanding hot combustion gas from combustion chamber. High turbine inlet temperature and lower exit temperature of the combustion

gas is desirable in order to achieve high thermal efficiency and work output. Due to the metallurgical constraints the turbine inlet temperature has to be maintained at a maximum permissible limit. In this study turbine inlet temperature is assumed to be 1273 K.

In order to determine turbine exit temperature the expansion of combustion gases assumed to be isentropic. The exhaust gas temperature (isentropic) of the turbine T_4 is estimated as:

$$\frac{T_3}{T_{4s}} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} \tag{3.28}$$

Alternate method to evaluate T_{4s}

Assuming isentropic expansion of combustion gas in the gas turbine, the entropy before and after expansion would be the same. So,

$$\Delta s = S_3 - S_{4s} = 0, \tag{3.29}$$

where,

$$S_3 = \left[s_{3,CO_2} + 2s_{3.H_2O} + (X-2)s_{3,O_2} + 3.76Xs_{3,N_2}\right] - R\left[\ln\left(\frac{1}{T_n}\right)\left(\frac{P_3}{P_0}\right) + 2\ln\left(\frac{2}{T_n}\right)\left(\frac{P_3}{P_0}\right) + (X-2)\ln\left(\left(\frac{X-2}{T_n}\right)\left(\frac{P_3}{P_0}\right)\right) + 3.76X\ln\left(\frac{3.76X}{T_n}\right)\left(\frac{P_3}{P_0}\right)\right] \tag{3.30}$$

and,

$$S_{4s} = \left[s_{4s,CO_2} + 2s_{4s.H_2O} + (X-2)s_{4s,O_2} + 3.76Xs_{4s,N_2}\right] - R\left[\ln\left(\frac{1}{T_n}\right)\left(\frac{P_3}{P_0}\right) + 2\ln\left(\frac{2}{T_n}\right)\left(\frac{P_3}{P_0}\right) + (X-2)\ln\left(\left(\frac{X-2}{T_n}\right)\left(\frac{P_3}{P_0}\right)\right) + 3.76X\ln\left(\frac{3.76X}{T_n}\right)\left(\frac{P_3}{P_0}\right)\right] \tag{3.31}$$

In the above Equations [3.30] and [3.31],

T_n - Total number of moles in products of combustion.

By solving the equation $\Delta s = S_3 - S_{4s} = 0$ the isentropic temperature T_{4s} can be found out.

In order to determine actual temperature at exit of the gas turbine following equation is used.

$$\eta_{turbine} = \left(\frac{T_3 - T_4}{T_3 - T_{4s}} \right) \tag{3.32}$$

Assuming isentropic gas turbine efficiency as 0.88; the actual temperature of the exhaust gas T_4 at the exit of the turbine is thus determined.

In energy analysis of gas turbine, mass, energy and entropy balance equations are developed. Referring to the figure 3.5 , the mass balance equation can developed by balancing mass of combustion gas entering into the gas turbine and the mass of combustion gases at the exit of the turbine. The equation is formulated as:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_g, \tag{3.33}$$

Where,

$\dot{m}_g$ -mass of combustion gas in kg/s

Further, the energy balance equation is developed for the gas turbine to establish relationship between enthalpy entering the system at state “3”; turbine work output and enthalpy of combustion gases at turbine exit. It is assumed that the turbine is leak-proof. Thus the energy balance across the gas turbine is formulated as:

$$\dot{m}_3 \; h_3 = \dot{m}_4 \; h_4 + W_T, \tag{3.34}$$

Alternatively the equation can be written as

$$W_T = \dot{m}_g \, c_{pg} \left(T_3 - T_4 \right), \tag{3.35}$$

where, W_T – Turbine work

The net-work output is the difference between the turbine output and compressor output. Thus, the net- work output can be expressed as:

$$W_{net} = \dot{m}_g c_{pg}(T_3 - T_4) - \dot{m}_a c_{pa}(T_2 - T_1) \tag{3.36}$$

In this case study, the net output of the turbine is 10MW. Thus the air, fuel and gas consumption are determined by solving the following equation.

$$W = \dot{m}_f \left(\left(1 + \frac{\dot{m}_a}{\dot{m}_f} \right) c_{pg}(T_3 - T_4) - \left(\frac{\dot{m}_a}{\dot{m}_f} \right) c_{pa}(T_2 - T_1) \right) \tag{3.37}$$

Referring to Figure 3.5; the change in entropy due to the expansion of combustion gases in the gas turbine is the difference between entropy at state "3" and state "4". The entropy balance equation across the gas turbine in BGCS is formulated as:

$$\dot{m}_3 \, s_3 = \dot{m}_4 \, s_4 - S_{genT} \tag{3.38}$$

The Equation 3.37 can be further simplified as:

$$S_{genT} = \dot{m}_g(\dot{s}_4 - \dot{s}_3), \tag{3.39}$$

The change in specific entropy at the state "3" and "4" can be determined by the following Equation [3.40] ([5]),

where, $$s_4 - s_3 = c_{pg} \ln\left(\frac{T_4}{T_3}\right) - \frac{k-1}{k}\left(\ln\left(\frac{p_4}{p_3}\right)\right) \tag{3.40}$$

And, s_3; s_4 are the specific entropies at given state.

The irreversibility in gas turbine (i_{turb}) can thus be evaluated by the following equation.

$$I_{turb} = T_0 S_{gen.T} \tag{3.41}$$

The exergy at the state "4" can be determined by evaluating exergy balance equation.

The exergy balance equation can be expressed as:

$$\dot{e}_{x3} = \dot{e}_{x4} - W_T + i_{turb}, \quad (3.42)$$

where, $\dot{e}_{x1}$ and $\dot{e}_{x2}$ are specific physical exergy at the given state 3&4.

3.4.4 Heat recovery steam generator in BGCS

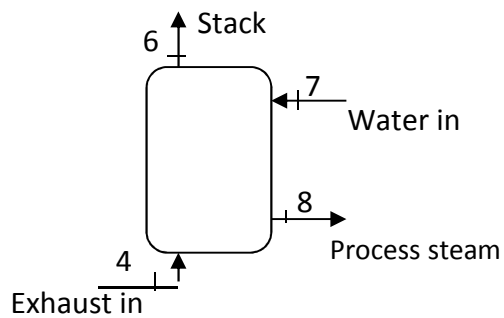


Figure 3.6. Schematic diagram of HRSG of the basic GT cogeneration system

In the HRSG calculations, the process steam rate is determined based on available exhaust gas enthalpy and steam injection mass flow rate. In addition, the saturated steam is generated by utilizing the heat energy from hot gas turbine exhaust gas. For this analysis, it was assumed that feed water with saturated temperature is fed in HRSG to obtain saturated steam.

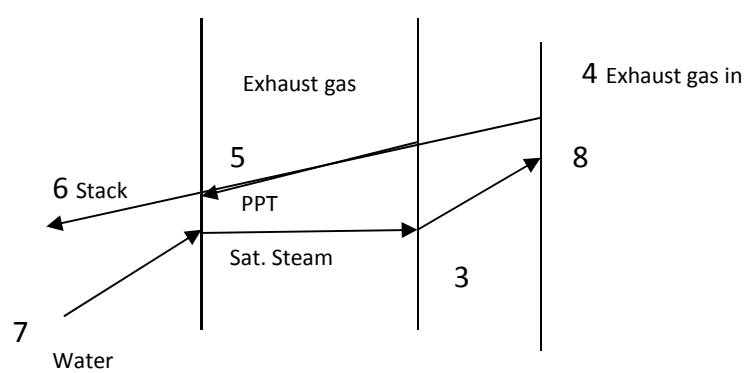


Figure 3.7. Temperature profiles across heat recovery steam generation in BGCS [17]

Referring to the figure 3.7, the exhaust gases from the gas turbine, enters at “4”. The heat energy from the exhaust gas is recovered by the water to produce saturated steam that can be used for process application. The exhaust gases with lower temperature then exhausted into the atmosphere through stack.

In the energy analysis of HRSG, mass and energy balance equations are developed. The mass flow of exhaust gases and inlet water entering into the HRSG are balanced by the mass flow of flue gases at the stack and saturated steam produced in the HRSG. It is assumed that the leakages in HRSG are negligible. Thus mass flow balance equation is written as:

$$\dot{m}_4 + \dot{m}_w = \dot{m}_6 + \dot{m}_s \tag{3.43}$$

The pinch point difference is assumed to be 20˚C. The process steam pressure is 10 bars. The saturation temperature of the steam is 180.875˚C. T_5 can be determined from Equation 3.44.

$$T_5 = T_{sat} + PPT \tag{3.44}$$

In a heat recovery steam generator, the energy flow balance equation is developed in order to determine rate of steam flow. Initially, the stack temperature T_6 is determined by solving the energy balance equation at T_5. The energy balance across HRSG is given as:

$$\dot{m}_g c_{pg} T_4 - c_{pg6} T_6 = \dot{m}_s (h_8 - h_7) \tag{3.45}$$

Thus the rate of the steam flow $\dot{m}_s$ for process heating can be determined. The heat energy generated in HRSG is evaluated by:

$$Q_{hrsg} = \dot{m}_g (c_{pg4} T_4 - c_{pg6} T_6) \tag{3.46}$$

In HRSG, the exergy flows at each state “4”,”5”,”6”,”7”and “8” are determined and the irreversibility are evaluated.

The exergy balance across HRSG was formulated as:

$$\dot{e}_{X4} - \dot{e}_{x6} = \dot{e}_{X8} - \dot{e}_{X7} - i_{rrev.HRSG} \quad , \tag{3.47}$$

where, exergy represents the exergy flow due to incoming exhaust gas in HRSG [21].

$$\dot{e}_{x4} = c_{pg4} T_0 \left(\left(\frac{T_4}{T_0} \right) - 1 - \ln \left(\frac{T_4}{T_0} \right) \right) \tag{3.48}$$

The exergy out is determined by [22]:

$$\dot{e}_{x6} = c_{pg4} T_0 \left(\left(\frac{T_6}{T_0} \right) - 1 - \ln \left(\frac{T_6}{T_0} \right) \right) \tag{3.49}$$

The exergy utilization rate is the exergy utilized to generate process steam is determined as [6]:

$$\dot{e}_{x8} - \dot{e}_{x7} = (h_8 - h_7) - T_0 (s_8 - s_7) \tag{3.50}$$

Solving Equation 3.47 irreversibility in the HRSG ($i_{rrevHRSG}$) is evaluated.

3.5 Steam Injected Gas Turbine Cogeneration System Analysis and Methodology

The basic idea of gas turbine humidification is that the injected steam increases the mass flow rate through the turbine. This increases the specific output as the compressor work remains constant. The enthalpy of the steam is utilized to increase the power. The heat energy from the exhaust is utilized to generate steam for injection and heating applications and the efficiency increases. For a given pressure ratio and TIT and a for constant air fuel ratio the required fuel quantity to generate fixed power decreases. Thus emissions are reduced. In the steam injected gas turbine systems, heat recovery steam generator operates on the pressure somewhat higher than the compressor discharge pressure. The steam is introduced in the combustor at the pressure higher than compressor discharge pressure. The mixture of steam and combustion products

passes into the turbine, where the augmented mass flow increases the power produced by the turbine. This power augmentation is due to the increased mass flow rate in the turbine. In the combustion chamber, the combustion of the fuel takes place with compressed air and high pressure steam. In the analysis, the steam is introduced in the combustion chamber but will not take any active part in the combustion process.

Beta, heat demand ratio, is the ratio of the heat quantity utilized for heating and cooling options to the maximum heat energy that can be used for the application. In this study as Beta increases that means more heat energy is extracted for the applications energy efficiency (power) decreases, but the heat generation efficiency increases. As a result of this the cogeneration efficiency increases. Since in the case studies the compressor remains the same, the properties of the state "2"will remain same.

3.5.1 Combustion chamber in STIG

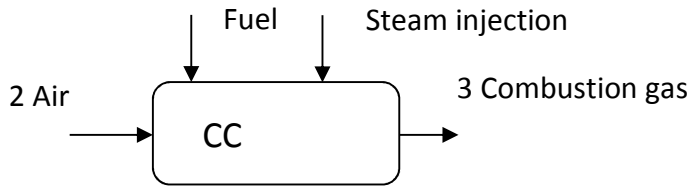


Figure 3.8 Schematic diagram of combustion chamber of steam injected cogeneration system.

In a steam injected gas turbine cogeneration system the combustion chamber is similar to the basic gas turbine cogeneration system except there is a provision for the steam injection. Referring to Figure 3.8, in the combustion chamber energy analysis the mass balance, the energy balance equations are developed as follows.

In coming mass flow of compressed hot air, fuel and injected steam in the combustion chamber is equal to the outgoing mass flow of gas after combustion. It is assumed that, there are no leakages in the STIG combustion chamber. The mass balance equation:

$$\dot{m}_a + \dot{m}_f + \dot{m}_s = \dot{m}_g \tag{3.51}$$

The combustion process is taken into consideration while developing energy balance equation. In case of steam injection, the combustion Equation 3.16 changes to

$$CH_4 + x[O_2 + 3.76N_2] + (s)H_2O \rightarrow (a)CO_2 + (b)O_2 + (c)N_2 + (d)H_2O \tag{3.52}$$

In STIG combustion chamber, high enthalpy steam is injected at higher pressure [5]. The energy balance across the combustion chamber is written as:

$$\dot{m}_a h_a + \dot{m}_f LCV + \dot{m}_s hs_{tinj} \rightarrow \left(\dot{m}_a + \dot{m}_f\right) h_g + \dot{m}_s h_s s_{T3} \tag{3.53}$$

In the calculations, the gas flows and steam flows are treated separately since the steam does not take part in combustion process. The quantity of the injected steam is so small that the pressure of steam injection is neglected.

In the case of steam injected gas turbine system, the exergy component of steam injection is added into the total exergy. The exergy at state "3" (e_{x3}) increases due to the steam injection, and increased mass flow rate as follows:

$$\dot{e}_{x2} + \dot{e}_{xfuel} + \dot{e}_{xinj} + i_{ccinj} = \dot{e}_{x3} \tag{3.54}$$

In the exergy analysis of STIG combustion chamber, Equation 3.21 and Equation 3.52 are used in order to determine exergy of reactants and products. The physical exergy due to the steam injection at turbine inlet state 3 is determined by following Equation [5]

$$e_{phsteame} = \left[\sum n_s (h_{T_3} - h_{298}) - T_0\left(\sum n_s \left(s_{T_3} - s_{298}\right) - n_s R \ln\left(\frac{p}{p_0}\right)\right)\right] \tag{3.55}$$

The irreversibility in the STIG combustion chamber (i_{ccinj}) is thus determined by Equation 3.54.

3.5.2 Gas Turbine in STIG

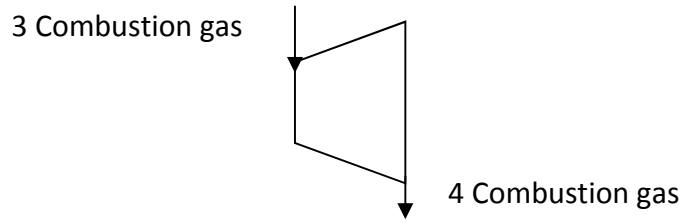


Figure 3.9 Schematic diagram of turbine for the steam injected cogeneration system

In STIG gas turbine the product of combustion is expanded from state "3" to state "4" as shown in Figure 3.9. The methodology used in developing mass, energy and entropy balance equations is similar to the methodology used in BGCS.

The combustion gas with high temperature and pressure enters the gas turbine at state "3" and the expanded combustion gas which has low pressure and temperature exits the gas turbine at "4". The mass balance equation is written as:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_g \tag{3.56}$$

The gross turbine power can be determined by carrying energy balance across the turbine.

$$W_T = (\dot{m}_a + \dot{m}_f + \dot{m}_s)(h_3 - h_4)_{gas} \tag{3.57}$$

Net power output at the gas turbine is determined by

$$W = \dot{m}_f \left(\left(1 + \frac{\dot{m}_a}{\dot{m}_f}\right) c_{pg} (T_3 - T_4) + \% \dot{m}_a h_s - \left(\frac{\dot{m}_a}{\dot{m}_f}\right) c_{pa} (T_2 - T_1) \right) \tag{3.58}$$

Any increase in net work output is due to the fact that, the turbine work increases due to steam injection, whereas work consumed by the compressor remains constant.

The entropy generated in the STIG gas turbine is determined by following equation:

$$\dot{m}_3 s_3 = \dot{m}_4 s_4 - S_{genT} \tag{3.59}$$

Equation 3.57 can be further simplified as:

$$S_{genT} = \dot{m}_{gs}(s_4 - s_3), \tag{3.60}$$

The change in specific entropy at the state "3" and "4" can be determined by Equation 3.39

Thus the irreversibility in STIG gas turbine (i_{turb}) can be evaluated by following equation.

$$i_{turbSTIG} = T_0 S_{gen.T} \tag{3.61}$$

3.5.3 Heat recovery steam generator in STIG

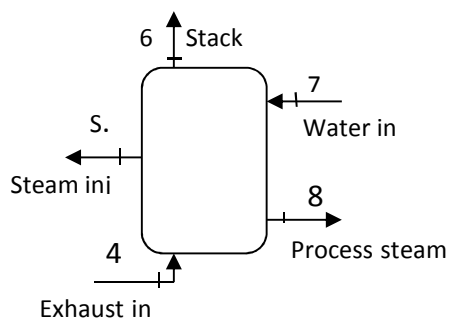


Figure 3.10. Schematic diagram of HRSG for the steam injected cogeneration system

In STIG, the steam generated in HRSG is preferentially used for steam injection and process heat. The part of the steam which is injected in the combustion chamber generates additional power. The pressure of the injected steam must be equal to or more than the combustion chamber pressure.

In energy analysis mass, energy and entropy balance equations are developed. Referring the figure 3.10 the mass balance equation is written as:

$$\dot{m}_{gs} + \dot{m}_w = \dot{m}_s + \dot{m}_{s.inj} + \dot{m}_{stack} \tag{3.62}$$

Energy balance equation is developed in HRSG in STIG in order to determine the rate of steam for the process application.

In HRSG, heat recovered from flue gas is determined as [9]:

$$\dot{m}_{gs} c_{pg} (T_4 - T_6) = \dot{m}_s (h_8 - h_7) + \dot{m}_{sinj} h_{sinj} \tag{3.63}$$

By solving Equation 3.63 the rate of process steam is determined.

The exergy destruction rate in HRSG is determined by evaluating the exergy balance equation similar to the BCGS.

The exergy utilization in the HRSG is determined by following equation:

$$\dot{e}_{x8} - \dot{e}_{x7} = (h_8 - h_7) - T_0(s_8 - s_7) + \dot{m}_{s.inj}((h_7 - h_6) - T_0(s_7 - s_6)) \quad (3.64)$$

3.5.4 Exergy Loss at the Stack

The stack temperature is controlled above dew point of the exhaust gas to prevent corrosion problems. The exhaust gas from the HRSG will go into the atmosphere through a chimney. The exergy of the exhaust gas at the stack temperature will be the exergy loss. It is determined by the using Equation 3.60.

$$\varepsilon_{phstack} = \left[\sum_k n_s\left(h_{T_8} - h_{298}\right) - T_0\left(\sum_k n_s\left(s_{T_8 - s_{298}}\right)\right) - n_s R \ln\left(\frac{p}{p_0}\right)\right] \quad (3.65)$$

The performance characteristics of STIG are evaluated by varying the steam injection rate. In this simulation study the steam injection is varied from zero to 0.1 percent of mass of air.

3.6 Performance Characteristics

The relationship among heat demand ratio and the performance characteristics is established. The heat demand ratio is the ratio of amount of process heat supplied (heat demand) to the maximum amount of process heat that can be utilized for the process application.

Heat demand ratio, beta (β) = $\frac{Q_s}{Q_{max}}$, (3.66)

where Q_s is the amount of heat demand and, Q_{max} is the maximum amount of process heat that can be utilized.

3.6.1 Specific work output

Having a high specific work output is positive. Because to get the same amount of power, less mass flow is required, thus the gas turbine becomes smaller.

The specific work output is calculated based on 1kg of gas generated in the combustion chamber. Applying mass and energy balance across combustion chamber following equation is developed to determine specific work output:

$$w = \dot{m}_f \left(\left(1 + \frac{\dot{m}_a}{\dot{m}_f} \right) c_{pg} (T_3 - T_4) - \left(\frac{\dot{m}_a}{\dot{m}_f} \right) c_{pa} (T_2 - T_1) \right) \tag{3.67}$$

The optimum pressure ratio is pressure ratio at maximum specific work output. The optimum pressure ratio is determined by:

$$r_{p_{opt}} = \left(\frac{T_2}{T_1} \right)^{\frac{k}{k-1}} \tag{3.68}$$

3.6.2 Thermal efficiency

Thermal efficiency of the gas turbine cogeneration system is the ratio of output in terms of power and generated heat to the fuel input. In this study, based on form of output, three thermal efficiencies are considered. The thermal efficiency based on power output (W_{net}); the power efficiency η_{power} is determined by following equation:

$$\eta_{power} = \frac{(W_{net})}{Q_{input}} \tag{3.69}$$

Based on combined heat and power output, the cogeneration efficiency

η_{cogen}, is determined by:

$$\eta_{cogen} = \frac{(W_{net} + Q_p)}{Q_{input}} \tag{3.70}$$

where $Q_{input} = \dot{m}_f \times CV$ (3.71)

Energy efficiency of the systems is calculated based on lower heating value of the fuel.

3.6.3 Exergetic efficiency

The exergetic efficiency is the ratio of exergetic output to the fuel exergy input. Three exergetic efficiencies based on three form outputs are determined. The exergetic efficiency $\eta_{powerexergy}$ based on power output W_{net} is determined by:

$$\eta_{powerexergy} = \left(\frac{W_{net}}{E_{xin}} \right) \tag{3.72}$$

The exergetic efficiency based on combined heat and power $\eta_{totalexergy}$ is determined as

$$\eta_{total\ exergy} = \frac{Exergy_{,}out}{Exergy_{,}in} = \left(1 - \frac{\sum irrev}{Ex_{in}} \right) \tag{3.73}$$

Exergy efficiency is evaluated based on exergy of the fuel. In this analysis the exergy of methane is assumed to be 51705kJ/kg.

3.6.4 CO_2 Emissions Analysis

Carbon dioxide emissions are estimated based on fuel consumption. Emissions are directly proportional to the fuel consumption. The quantity of carbon dioxide in moles produced during combustion of methane is determined by the combustion chemical Equation 3.16. Number of moles then converted to kg by multiplying the ratio of molecular weight of CO_2 to the molecular weight of CH_4.

$$CO_2 emissions\_kg/s = \frac{b_1 (M_{CO_2})}{M_{CH_4}} \quad , \tag{3.74}$$

where b_1 is the coefficient of CO_2 in combustion equation

Similarly, specific CO_2 emissions in terms of kg / kWh are evaluated by the following Equation (3.75 [13])

$$CO_2 emissions\_kg/kWh = \frac{3600 \times m_f \times M_{CO_2}}{M_{CH_4} \times W_{net}} \tag{3.75}$$

3.7 Effect of Variation in Specific Heat as a Function of Temperature

The specific heat is defined as the amount of heat energy gained or lost per unit mass of substance to raise its temperature by one degree Celsius or Kelvin. Earlier, the specific heat was taken as constant throughout the cycle. Due to this assumption the considerable error is encountered in the analysis. This is due to the fact that difference in specific heat value between cold air and hot gases. In fact, the specific heat of air and the combustible gases always changes during compression and expansion processes. Besides that, the product of combustion gas contains primarily CO_2 and H_2O which have high specific heat than pure air.

The specific heat, a physical property of the material is calculated by taking sum of the products of mole fraction times the specific heat of that gas component.

For BGCS, the chemical equation of the combustion is as follows:

$$CH_4 + X\,(O_2 + 3.76\,N_2) = b_1CO_2 + b_2\,H_2O + b_3O_2 + b_4N_2 \qquad (3.76)$$

For system Configuration II, due to steam injection the chemical reaction changes to

$$CH_4 + X\,(O_2 + 3.76\,N_2) + a1{*}H_2O = b_1CO_2 + b_2\,H_2O + b_3O_2 + b_4N_2 \qquad (3.77)$$

The mean specific heat per mole of individual combustion gas constituents is taken from Thermodynamic standard tables [6]. The coefficients b_1, b_2, b_3 & b_4 are taken from gas property tables [5]

$$c_{p_{meangas}} = a + bT + cT^2 + dT^3, \qquad (3.78)$$

where 'a' is the coefficient of CO_2, 'b' is the coefficient of H_2O, 'c' is the coefficient of O_2 and 'd' is the coefficient of N_2.

3.8 Range of Operating Conditions

The operating conditions considered in this study are outlined in the table below. To analyze the performance the small/medium range gas turbine cogeneration system 10

MW is chosen. The small /medium range cogeneration systems are common and cover wide applications [6]. The simulation results can be projected either way (maximum or minimum) to estimate the performance of gas turbine cogeneration systems. For reference basic information such as type of fuel steam condition at HRSG with regards to the operating conditions is taken from the simulation study conducted by Pak and Suzuki [12] on the similar gas turbine cogeneration systems. The range of pressure ratio is chosen as 8-22. The typical range of gas turbines for industrial applications is up to 18 [6]. The range of turbine inlet temperature is taken as 1173K -1873K. The gas turbines run on turbine inlet temperature up to 1673K. The performance of gas turbine cogeneration system also investigated for higher pressure ratios 20, 22 and turbine inlet temperatures 1773 and 1873 K for the study purpose.

Table 3.1 Operating conditions of gas turbine cogeneration systems [12]

1	Ambient conditions	1 bar and 25°C
2	Fuel	Methane
3	Fuel gas conditions	1 bar and 25°C
4	Fuel Gas composition	100% Methane
5	Pressure ratio, range	8-22
6	Turbine Inlet temperature, range	1173K-1873 K
7	Net Power output	10 MW
8	Water inlet condition	1 bar ,100°C
9	Steam condition for the heat supply	Saturated ,1 bar
10	Steam condition for the steam injection	Superheated ,
11	Specific heat of the Air	1.006 kJ/kg K
12	Specific heat ratio	1.4
13	Standard chemical exergy of the fuel[5]	836510 kJ/k mole

3.9 Gas Turbine Cogeneration Systems Simulation Methodology

In this study the performance analysis of gas turbine cogeneration systems is assessed. The work presented has been conceptualized and developed through step by step approach. The analysis started with understanding the operating conditions. Each component of gas turbine cogeneration is then modeled. The compressor is first considered for the analysis. The actual temperature at the compressor outlet is determined by evaluating for the isentropic temperature in compressor. The isentropic efficiency is assumed to be 0.85 as mentioned earlier. The air/fuel ratio is evaluated by solving the energy balance Equation 3.14, across the combustion chamber, considering 1 kg of combustion gas. Assuming complete combustion in combustion chamber, the amount of excess air is determined. The turbine inlet temperature is maintained at 1273K by changing the air fuel ratio. The actual temperature at the turbine exhaust can then be determined assuming the isentropic turbine efficiency to 0.85. In this analysis the air fuel ratio is varied to maintain constant power output of 10MW. By keeping fixed power output the influence of varying operating variables on the performance of gas turbine cogeneration systems is evaluated.

The actual amount of fuel, air and generated combustion gas is determined by a derived Equations 3.37 and 3.58, from energy balance across combustion chamber. The specific heat of the products of combustion at each state (Stages 3 to 6) is determined. The combustion Equations 3.16 and 3.52 are balanced considering 1 Kg mole of methane and mole fractions of the products of combustion are then determined by equating the coefficients of reactants and products. These mole fractions are used to determine the specific heat of each element of the product of combustion. With the help of Equation 3.78 the specific heat of combustion gas can be determined.

The temperature profile is used to determine the mass of steam generated in the HRSG. The temperature at the stack is determined by making energy balance at pinch point.

The steam injection effect is observed by varying heat demand ratio. In the analysis the quantity of steam is taken as percentage of amount of air used in combustion process.

In exergy analysis the irreversibilities of each component of gas turbine cogeneration system is determined. In case of compressor and turbine the change in entropy are evaluated in order to determine irreversibilities physical and chemical exergy of product and reactants are considered. In HRSG, the irreversibilities are determined by taking the difference between exergy inflow and exergy out flow.

The CO2 emissions of both the gas turbine cogeneration systems are estimated with reference to unit power output and per unit cogeneration output using Equation 3.75.

In order to investigate the influence of specific heat on the performance parameters the calculations are performed by considering constant specific heat and as that of air (air standard cycle). The performance results are compared with the results obtained by constant specific heat (air standard).

In this study all the known properties, specific heat, mass, energy and entropy balance equations are integrated using Microsoft Excel. The calculation flow charts are given in Appendix A1 and Appendix A2. The performance of both gas turbine cogeneration systems is evaluated based on the performance parameters. A comparison between the two gas turbine cogeneration systems, without steam injection and with steam injection is presented.

CHAPTER 4. RESULTS AND DISCUSSIONS

Methodology which was discussed above is used to analyze the cogeneration systems to evaluate the performance of the basic gas turbine cogeneration system (BGCS) and steam injected gas turbine cogeneration system (STIG). In the thermodynamic analysis, each component in the system was modeled. The mass, energy and entropy balance equations were formulated to determine the properties of each state in the systems. A flow chart was developed to help understand the steps in the calculation process. Each stage was described by the input and output conditions. In the case studies, turbine power output was constant at 10 MW. In this systems, the influence of key operating variables such as heat demand ratio, pressure ratio and turbine inlet temperature (TIT) on specific work output, energy and exergy efficiencies were investigated. In the exergy analysis, component wise exergy destruction/losses were evaluated to determine distribution of the exergy destruction in the systems.

In the carbon dioxide emission analysis, the carbon dioxide produced during the combustion was estimated for both cogeneration systems. Effects of pressure ratio, TIT and heat demand ratio on CO_2 emissions were discussed. The results show that the steam injection reduces the CO_2 emissions.

Methane was used as the fuel in the system. Taking the chemical composition and temperature variation of the gases into consideration the specific heat of the reactants and the products in the combustion process was evaluated, the performance parameters were estimated based on the formula given in Moron and Shapiro [6]. The specific heat is one of the major influencing factors in the performance calculations of any thermal system. This has been demonstrated by performing calculations by taking constant specific heat that of air into account and the results were compared. Significant variation was found in the results and this pose a risk of error in estimating the cogeneration system performance. The effects of heat demand ratio and the steam injection were also demonstrated.

In the parametric studies, pressure ratios r_p from 8 to 22 and turbine inlet temperatures from 1173K to 1873K were selected for the performance analysis.

4.1 Effect of Heat Demand Ratio on Performance of GT Cogeneration Systems

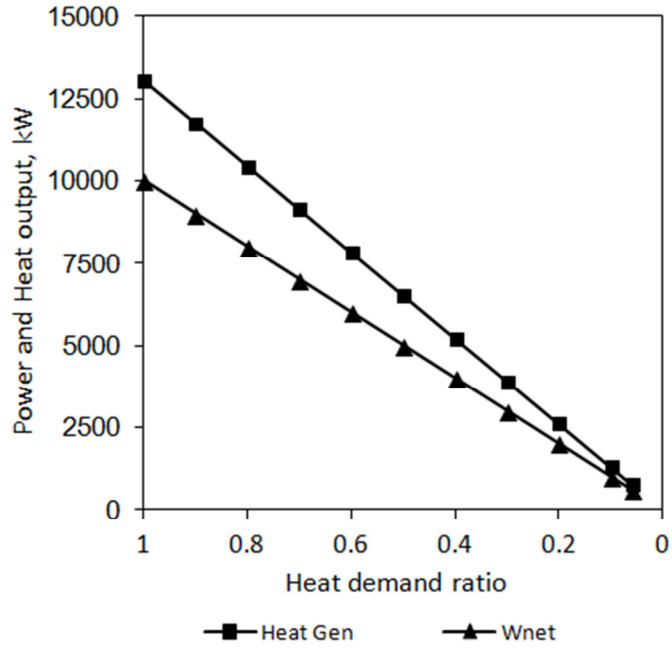
One of the important considerations when choosing appropriate cogeneration systems is to understand the thermal energy needs and electrical demand. The cogeneration plant should be capable of meeting total steam and electrical requirements. So it is necessary to understand how particular cogeneration system performs under changing heat demand. The heat demand ratio defined as the heat energy demand to the maximum heat energy that can be supplied to the process application. In this analysis effect of change in heat demand ratio on heat generation and power output was investigated.

4.1.1 Basic gas turbine cogeneration system (BGCS)

The basic gas turbine cogeneration system comprises a compressor, a combustion chamber, a turbine and HRSG. In this system, fuel consumption depends on required process heat and power output. The highest rating net power output was 10 MW. The maximum process output was about 13 MW. As the heat demand decreased, the fuel consumption decreased accordingly and the power output was thus reduced. The Figure,4.1 shows the estimated power output and generated heat in basic gas turbine cogeneration system when heat demand was changed. The heat demand ratio is plotted on horizontal axis in opposite direction and on vertical axis generated power and heat. The heat demand ratio decreased from 1 to 0. It can be seen that the net power output of turbine decreased significantly with decrease in heat demand ratio.

This is an efficient and economic system for applications with high heat loads. However, when the thermal load drops the operating economics are penalized severely. This is

because as the heat demand lowers, the generated power goes down, since the rate of fuel consumption lowers.



Figure,4.1. Effect of heat demand ratio on power output and generated heat in BGCS (r_p =8; TIT= 1273K)

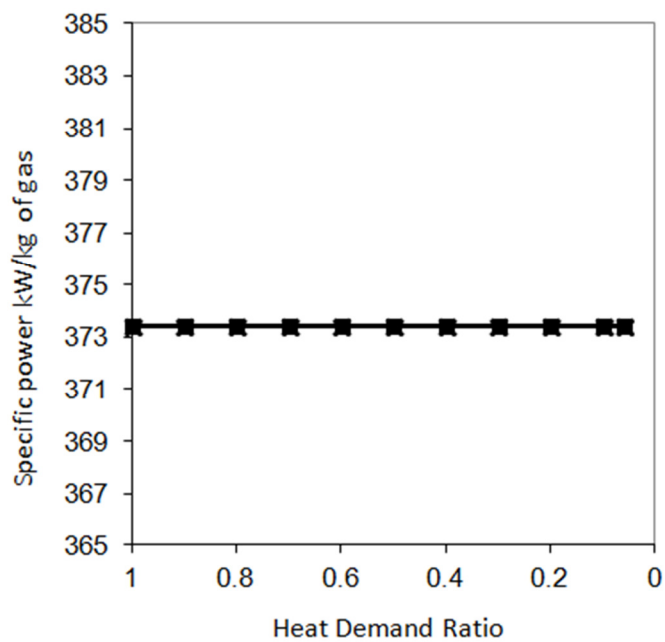


Figure 4.2. Effect of heat demand ratio on specific work in BGCS (r_p = 8; TIT= 1273K)

Figure 4.2 shows the effect of heat demand ratio on specific work output, energy and exergy efficiencies. The specific work output is defined as the power output per unit combustion gas. In the BGCS as the heat demand ratio decreased the mass of combustion gas also decreased due to lesser fuel consumption. Thus the specific work output remains constant. At given conditions (pressure ratio 8 and turbine inlet temperature 1273K) the specific work comes out to be 373.44 kW/ kg.

By definition the thermal efficiency is the ratio of power output and input (fuel consumption). In this case, as the heat demand decreases, the fuel consumption also decreases. So the efficiencies of the system remain constant. For instance in thermal efficiency at given conditions were found to be 36.15% whereas the exergetic power efficiency was found to be 35% as shown in Figure 4.3.Thus specific work output and efficiencies are insensitive to variation in heat demand ratio.

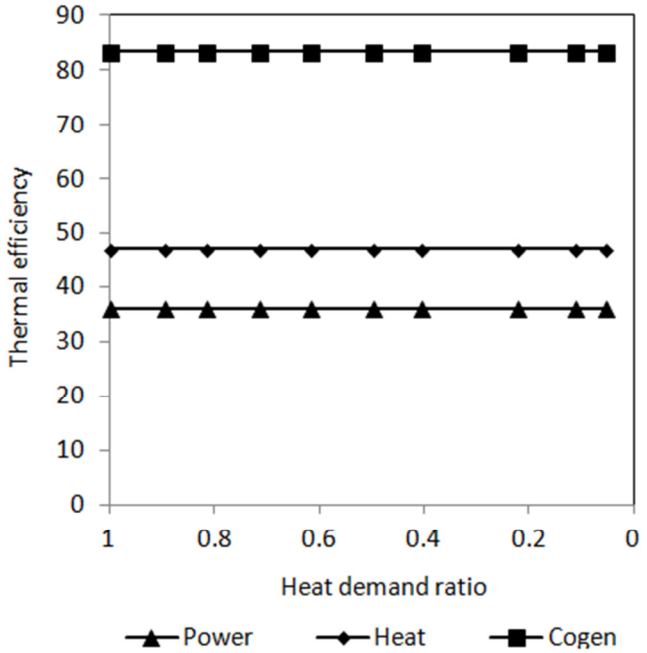


Figure 4.3 Effect of heat demand ratio on thermal efficiencies in BGCS (r_p =8; TIT =1273K)

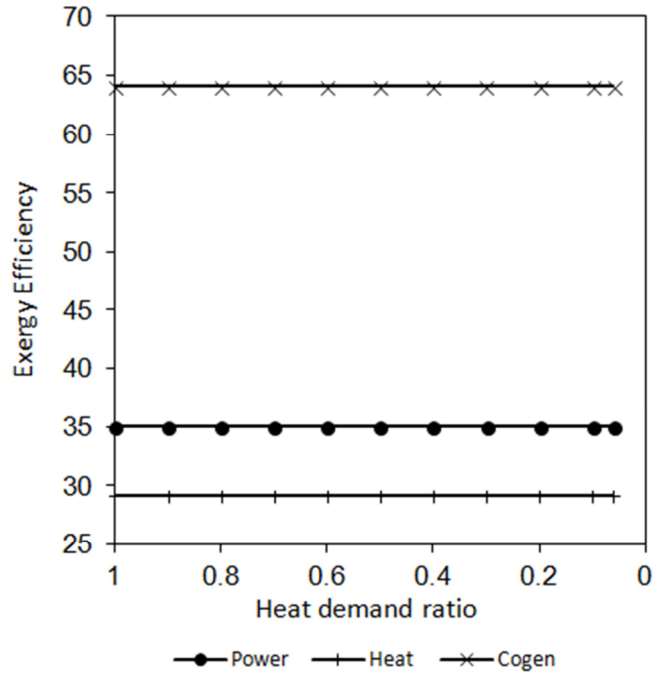


Figure 4.4 Effect of heat demand ratio on exergetic efficiency in BGCS (r_p = 8; TIT= 1273K)

In the basic gas turbine cogeneration system, when heat loads do fluctuate below rated output, two options are available in practice [36]:

a) Reduce the steam output or de-rate the gas turbine and reduce both steam output and power output.

b) Bypass the exhaust around boiler.

High electrical prices relative to fuel prices favour maintaining electrical output whereas lower of this ratio favours the de-rating. However bypassing the exhaust steam represents a waste of potentially useful energy and de-rating means lowering the output of capital investment.

4.1.2 Steam injected gas turbine cogeneration system

In the steam injected gas turbine cogeneration system, the generated steam in HRSG is introduced into the combustor to increase power output when heat demand is small. In this system the steam was utilized preferentially for application purpose and the remaining steam is injected to raise the power. Figure 4.5 shows the net power output

and generated heat output at constant turbine inlet temperature (1273 K) and various levels of heat demand ratios. It should be noted that the characteristics of the steam injected system are the same as that of the basic GT cogeneration system when there is no steam injection. The net power output 10 MW and 13 MW of process steam. The steam generated in the HRSG is utilized for process application. As the heat demand ratio decreased, the residual steam in the HRSG is utilized to inject steam into the combustion chamber. Thus, the heat demand ratio is inversely proportional to the steam injection. As the heat demand ratio decreased the power output increased as a result of steam injection and quantity of heat generated decreased. In this study, the lowest heat demand ratio considered was 0.06. At this point the power output was augmented from 10 MW to 14.8 MW. The minimum amount of steam that was available for process heat is less than 0.75 MW, while the maximum quantity of steam injection was found to be 4.45 kg/s. The minimum heat demand ratio was equal to 0.06. The heat demand ratio cannot be made zero that is because certain temperature difference is required to maintain between exhaust gas and steam generated in HRSG.

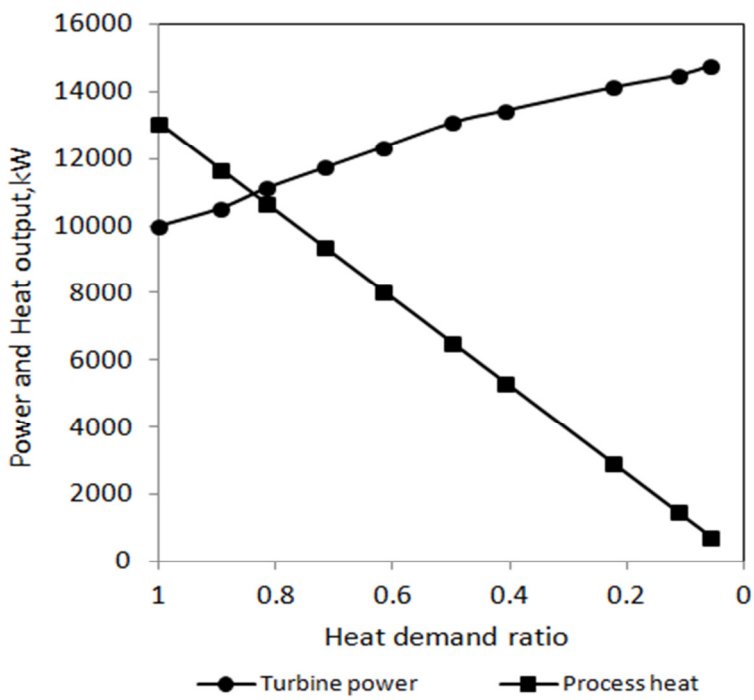


Figure 4.5 Effect of heat demand ratio on work and heat output in STIG (r_p = 8; TIT= 1273K)

The specific power is defined as; net-work output per unit mass of gas after combustion. As the Figure 4.6 shows, the specific work output increased with decreasing heat demand ratio. For a given pressure ratio the specific power increased with increasing steam injection ratio because the mass flow rate in the turbine increased, due to steam injection whereas that in the compressor mass flow rate is fixed. When heat demand is low, the excess steam can be injected into the turbine to increase the power output. The specific work increased from 373.44kW/kg to 390 kW/kg of gas, as the heat demand ratio decreases from 1 to 0.06.

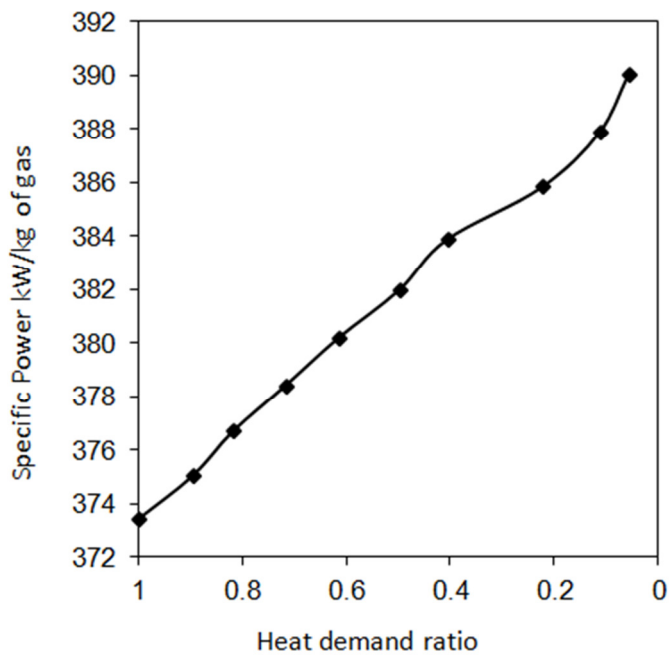


Figure 4.6 Effect of heat demand ratio on specific power in STIG (r_p = 8, TIT =1273 K)

The Figure 4.7 shows the effect of heat demand ratio on power, heat and cogeneration efficiency. In some applications, extra steam is generated for steam injection in order to increase work output, and at the same time to provide heat demand. At low heat demand the excess steam is utilized to increase the turbine power. As the heat demand ratio decreases the power generation efficiency increased. The power efficiency increased from 35% to 53%, however the cogeneration efficiency decreases from 82% to

55%, with increase in steam injection due to a drop in heat generation efficiency from 47% to 10%. In comparison with the basic gas turbine cogeneration system the drop in cogeneration efficiency was less when heat demand lowers. Thus steam injection system was more efficient than the basic gas turbine cogeneration system with regard to handling heat load fluctuations.

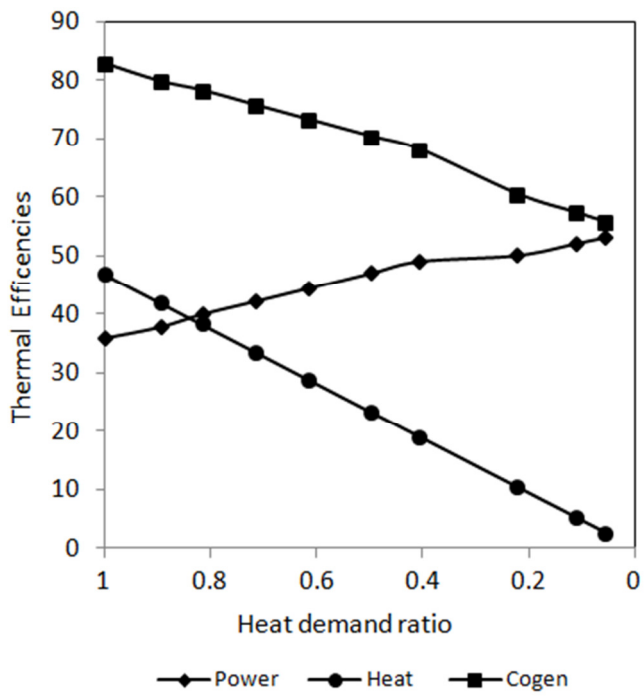


Figure 4.7 Effect of heat demand ratio on thermal efficiencies in STIG (r_p = 8, TIT= 1273 K)

Figure 4.8 shows that total exergetic efficiency decreases with deceasing heat demand ratio. The total exergetic efficiency decreased from 64 % to 52 %. This is because of the combined effect of power and heat generation exergy. Due to the steam injection the exergy of power generation increased however the exergy of heat generation decreases. The net effect is a decrease in total exergetic efficiency which decreased from 63 % to 51% with decrease in heat supply ratio. Even though exergy efficiency of heat generation reduced from 28 % to 0.07%, exergy of power generation increased from 35% to 51%. The drop in exergetic efficiency due to lower heat demand is lesser than

the drop in basic gas turbine cogeneration system. Thus higher exergetic efficiency can be obtained by steam injection compared to basic gas turbine cogeneration system.

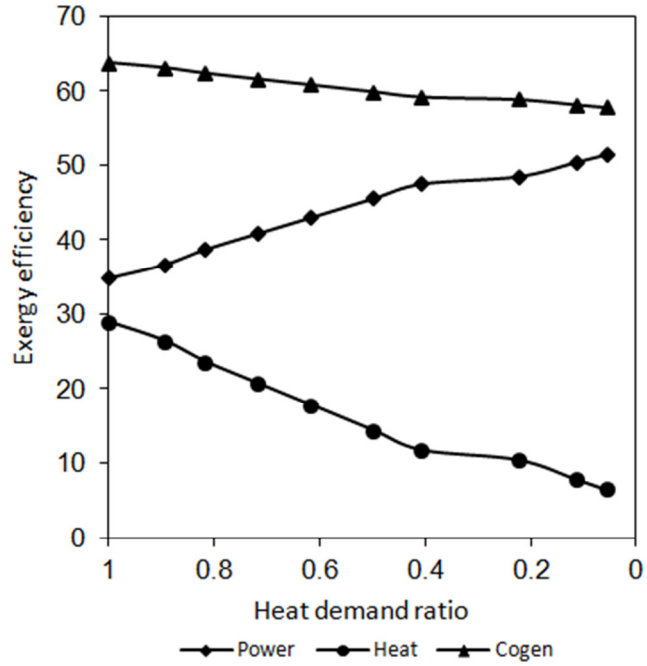


Figure 4.8 Effect of heat demand ratio on exergetic efficiencies in STIG (r_p = 8, TIT =1273 K)

Thus this system is beneficial where there is fluctuation in heat demand; unlike the exergetic efficiency it is less sensitive to the fluctuation in heat demand. In this system, the higher the heat supply ratio, the higher the exergetic efficiency. In comparison with the basic gas turbine cogeneration system the exergetic efficiency decreased to a lesser extent as the mass of steam injection increased. The steam injection system offers flexibility in operation and operate**s** in economical manner. It provides the choices to operate the cogeneration system depending upon load and energy price characteristics. There are the following three common choices to operate the cogeneration system.

a. The specific operating point is determined by process heat demand and electricity power output is allowed to float.
b. In the situation when the electricity prices are high, the economics dictate the system to run on maximum electrical output.

c. In the situation when the electricity buy back rate is too low, the power and heat demand is matched simultaneously to meet both requirements.

4.2 Effect of Pressure Ratio on Performance of GT Cogeneration Systems

Pressure ratio is one of the key operating variables that affect performance of gas turbine cogeneration system. It is seen that the thermal efficiency of the gas turbine cogeneration system can be increased as the pressure ratio are varied. This is demonstrated by varying pressure ratios and analyzing the performance by varying pressure ratios. In this analysis the range of pressure ratios is chosen from 8 to 22.The turbine inlet temperature is fixed at 1273K. The effect of the pressure ratio on performance characteristics such as specific work output, energy and exergy efficiencies are examined in this analysis.

4.2.1 Effect of pressure ratio on specific work output

The specific power is defined as net work output per unit mass of combustion gas. Figure 4.9 shows the relationship between pressure ratio and specific work output. The specific work output increased with increase in the pressure ratio up to its max value (378.63 kW/ kg) and then dropped. This is due to fact that at high pressure ratio, the temperature of the air at the compressor outlet increased which in turn reduced fuel consumption for a fixed turbine inlet temperature. Another reason for high efficiency and specific output is increase in mass flow rate for the higher pressure ratios. The specific work output increased up to a certain point due to the limitation in achieving turbine inlet temperature. This limit causes lower specific work output of the gas turbine cogeneration system for higher pressure ratios. To obtain same amount of work output, for higher pressure ratios, greater amount of air mass flow rate is required which calls for bigger and expensive compressor requirements. The network output is reduced, as a chunk of the power generated by turbine is consumed by a massive compressor which results in lower network output. The figure also points out the effect of steam injection. In the STIG as the heat demand ratio decreases the excess steam is

injected to augment the power. Due to the power augmentation the specific power increased with decreasing steam injection for a given pressure ratio. It is desirable to have high specific work output. Having high specific work output means, to get same amount of power, the mass of gas required is less hence smaller gas turbine.

In this analysis, the maximum pressure ratio was found to be 10. As the heat demand ratio decreased the specific output for the optimum ratio increased from 378.63 to 395.7 kW/kg of gas.

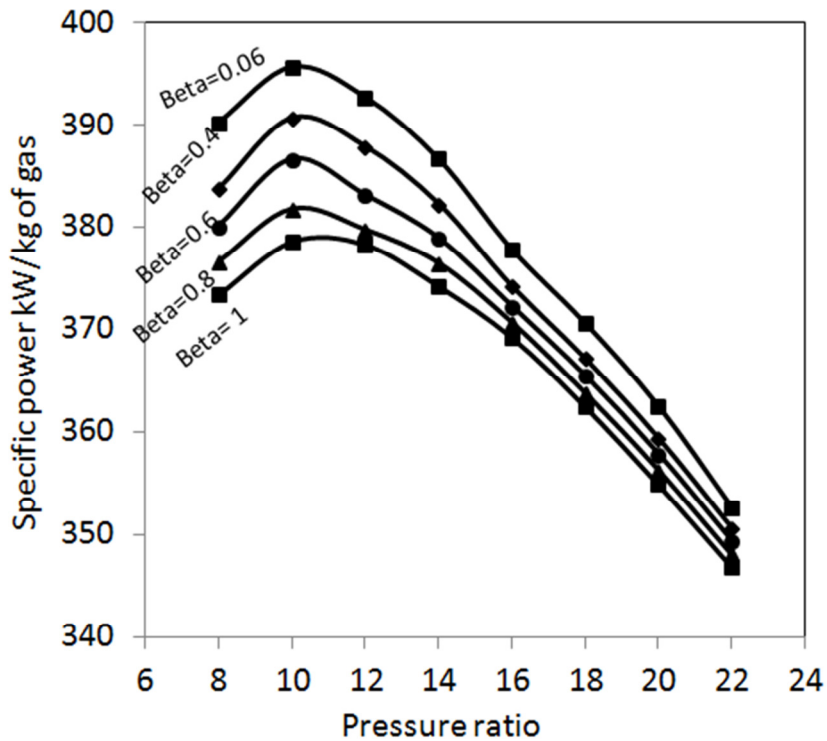
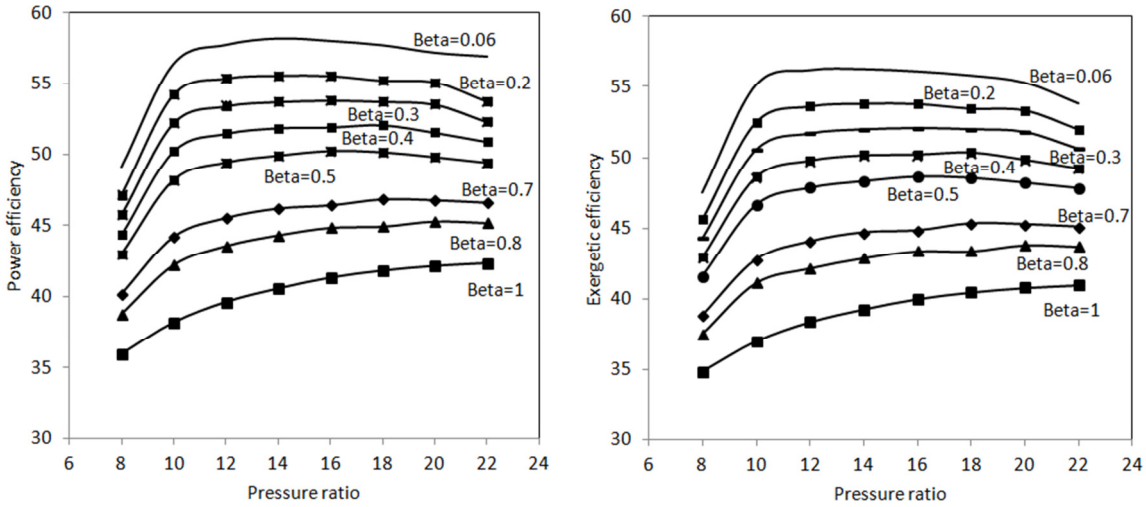


Figure 4.9. Effect of pressure ratio on specific work output in GTCS (TIT= 1273K)

4.2.2 Effect of pressure ratio on energy and exergy efficiencies

Figure 4.10 shows the relationship between pressure ratio and thermal efficiencies. The power efficiency increased with increase in pressure ratio. Although the power efficiency is proportional to the pressure ratio, it was observed that the rate of increase in power efficiency decreased for higher pressure ratios. In the BGCS the power efficiency increased from 36% to 42.3%.

Figure 4.10 also depicts the effect of steam injection. It is shown that for given pressure ratio, the power efficiency increased with a decrease in demand ratio. When the heat demand ratio was one, in absence of steam injection, the system behaved as the basic gas turbine cogeneration system. When the heat demand ratio decreased, the excess steam injected into the combustor to augment the power. In this study, for instance at pressure ratio 8 the power efficiency increased from 36% to 49.11%. This is due to reduced heat demand ratio from 1 to 0.6; the excess steam is injected in the combustor. Due to increased mass flow, greater power was generated. In the STIG, like the BCGS the rate of increase in power slowed down though, at higher pressure ratios.



(a) (b)

Figure 4.10 Effect of pressure ratio on power efficiency in GTCS (TIT = 1273K)

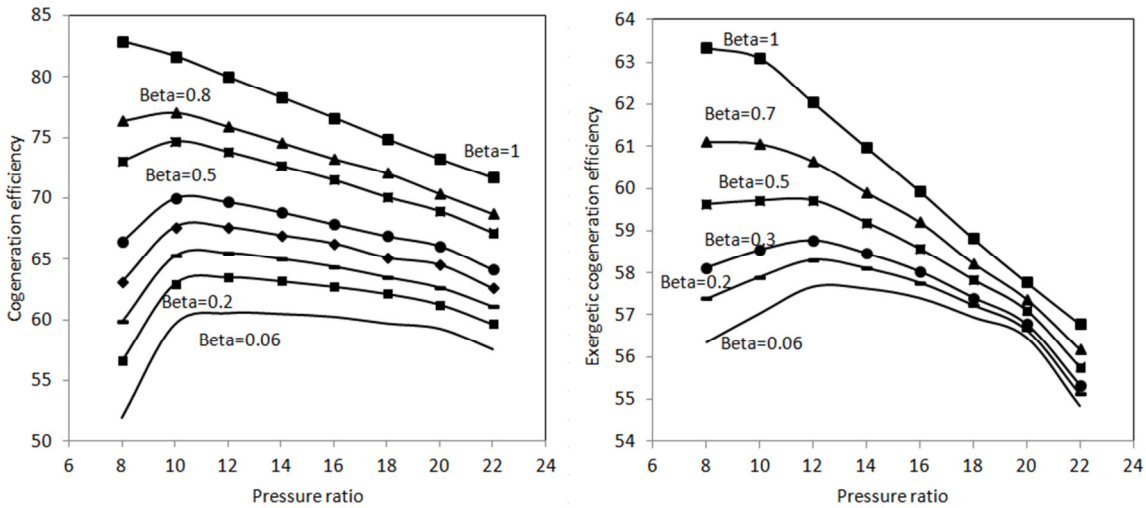
The power exergetic efficiency also increased with increase in pressure ratio as exergetic power efficiency is the function of work output. The work output increased with increase in pressure ratio. The power exergy efficiency increased from 34.8 to 47.51% in the BGCS. Like power thermal efficiency, in STIG the exergetic power efficiency also increased with pressure ratio due to power augmentation effect. It is observed that the power exergetic efficiency increased from 34.8 to 47.5%.

Figure,4.11 represents the effect of pressure ratio on cogeneration efficiency. It represents the combined effect of heat and power output. The inclination of the trend is decided by the influential part of the output (power and heat). In this case study, it is shown that when the pressure ratio increased, the cogeneration efficiency decreased. This is due to decreased heat generation efficiency at higher pressure ratios. The reason being that at higher pressure ratios, the gas turbine outlet temperature (T_4) decreased. As a result, the heat energy in the combustion products available to convert into steam is also reduced. It can also be seen that, for a given pressure ratio the heat generation efficiency was decreased with decrease in heat demand ratio. As a result the net effect of power and heat efficiency, of the BGCS decreased from 83% to 71.7 %.

For STIG, the same trend was observed. In the STIG, the cogeneration efficiency decreased at the higher pressure ratios. Though there was increasing trend in power efficiency, it is observed that when pressure ratio increased, the heat generation efficiency decreased. In addition, it is seen that, when heat demand ratio decreased, the quantity of steam generated for process applications also decreased which resulted in decreased heat generation efficiency. It is also important to note that, when the heat demand ratio decreased, the calorific value of the gas mixture reduced due to steam injection which in turn reduced inlet temperature. To maintain turbine inlet temperature, constant additional fuel is required to raise the calorific value of the gas mixture. Thus the efficiency dropped further. It is revealed that, in the STIG, at pressure ratio of 8,and TIT 1273K the cogeneration efficiency decreased from 83% to 52%.

Similar to the thermal cogeneration efficiency, the exergetic cogeneration efficiency also represents the net effect of exergetic power and exergetic heat generation efficiency. The net effect of power and heat exergetic efficiency is reflected in the exergetic cogeneration efficiency. It is found that the cogeneration exergetic efficiency decreased with increasing pressure ratios. The downward trend of the cogeneration exergetic

efficiency is due to decreased heat generation exergetic efficiency. It is observed that due to the steam injection the turbine exhaust temperature decreased which resulted in reduction of heat generation exergetic efficiency. In the analysis, exergetic heat generation efficiency in BGCS found to be decreased. So the exergetic cogeneration efficiency of BGCS reduced from 63.3 % to 56.7%, when pressure ratios increased from 8 to 22.



(a) (b)

Figure,4.11. Effect of pressure ratio on cogeneration efficiency in GTCS (TIT = 1273K)

While in STIG, it is observed that the exergetic efficiency of heat generation decreased from 29% to 18 %. This resulted in reduction of cogeneration exergetic efficiency from 63.7% to 56.3%.

4.3 Effect of Turbine Inlet Temperature on Performance of GT Cogeneration Systems

4.3.1 Effect of turbine inlet temperature on specific work

Turbine inlet temperature is another key operating variable in a cogeneration system. The efficiency and net work output of a gas turbine cogeneration system increased with increase in turbine inlet temperature. A high turbine inlet temperature also allows a gas

turbine cogeneration system for higher pressure ratio to improve the efficiency and maintains high net work output. There are metallurgical constraints to achieve high turbine inlet temperature. At high temperature the blades of the turbine start failing which reduces its durability and life [9]. In this work the range of temperature is considered from 1173K to 1873K.

At a higher turbine inlet temperature the enthalpy of the gases is higher. At higher temperatures, the specific heat of gases is also more. This results in a higher work output for fixed mass flow rate through gas turbine. Figure 4.12 (a) shows the relationship between TIT and specific power in basic gas turbine cogeneration system. The specific work output increased with increase in turbine inlet temperature. In the BGCS the specific power rose from 303.5kW/kg at TIT 1173 K to 753kW /kg at TIT 1173 K; and pressure ratio 8. In this case the pressure ratio is fixed to 8.

Figure 4.12 (b) represents the effect of turbine inlet temperature on specific work in the STIG. For a given heat demand ratio, the specific work increased with rise in turbine inlet temperature, due to steam injection.

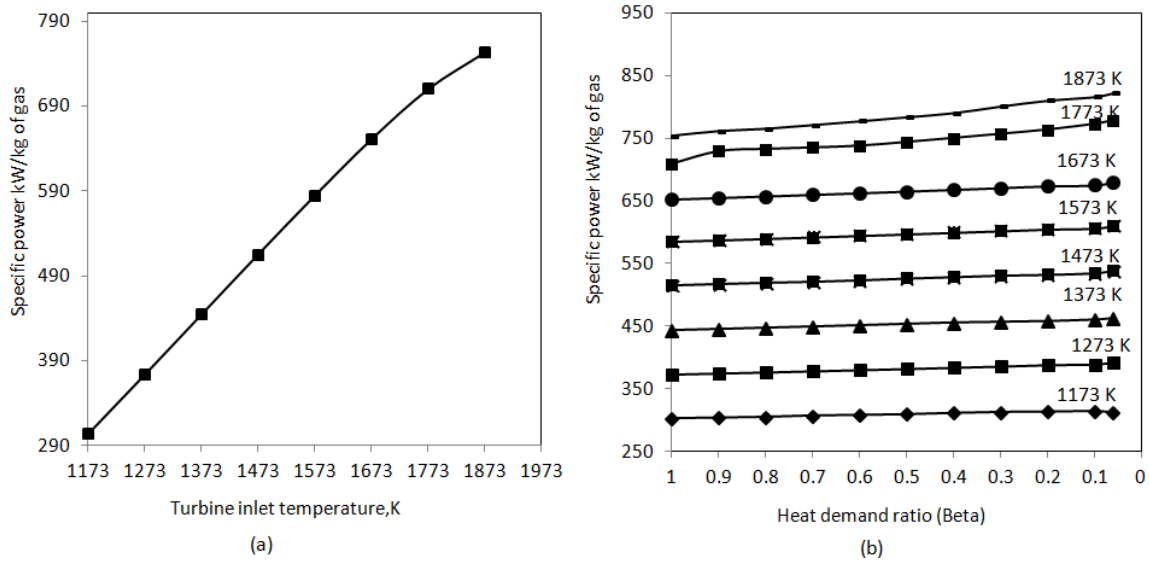
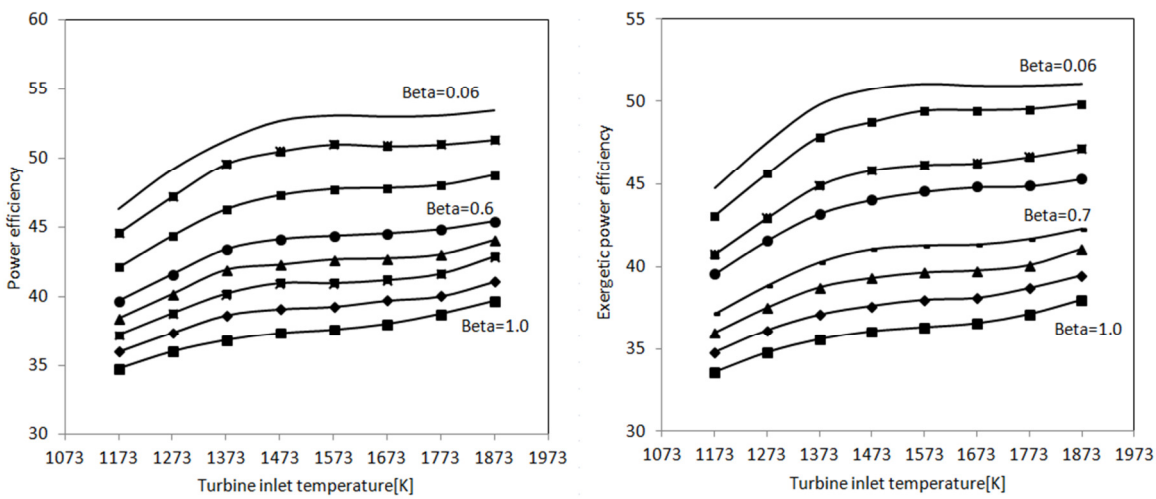


Figure 4.12 Effect of turbine inlet temperature on the specific work output in GTCS (r_p = 8)

For instance, at pressure ratio 8 and heat demand ratio 0.6, the specific power increased from 308.6 kW/kg of gas at 1173 K to 776.74 kW/kg of gas at 1873 K. In the STIG the escalation of specific power can be attributed to three factors, first one due to increased enthalpy at elevated turbine inlet temperature (21067 kJ/kg at 1173 K to 43928 kJ/kg at 1873K, second one due to specific heat of gas mixture (1.254 kJ/kg K; at 1173K to 1.393 kJ/kg K; at 1873K) and third one due to increased mass flow rate through the turbine.

4.3.2 Effect of turbine inlet temperature on energy and exergy efficiency

Both energy and exergy efficiencies increase with increase in turbine inlet temperature. Figure 4.13 describes the relationship between the turbine inlet temperature and power efficiencies. In this study, the turbine inlet temperature was varied from 1173K to 1873K. In the basic gas turbine cogeneration system, at pressure ratio 8, the power efficiency increased from 35 to 39 %. This is due to higher enthalpy and specific heat of gas mixture at increased turbine inlet temperature, the. Alike, energy efficiency the exergetic power efficiency is proportional to the turbine inlet temperature, also increased from 33 % to 38%.

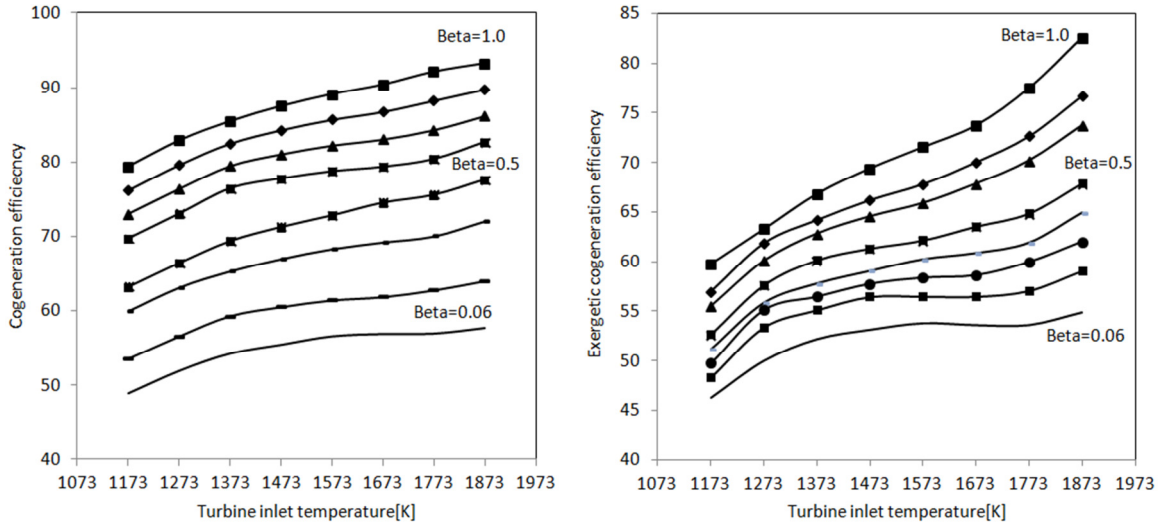


(a) (b)

Figure 4.13 Effect of gas turbine inlet temperature on power efficiency in GTCS (r_p = 8)

Figure 4.13 (a) also shows the steam injection effect. In the STIG the power energy efficiency increased with steam injection at higher turbine inlet temperatures. For instance at r_p 8 and TIT 1173K the energy efficiency was found to increase from 34.8% to 46.3% at fixed turbine inlet temperature. The exergetic power efficiency followed the same trend as that of power energy efficiency. The exergetic power efficiency increased from 33.6% to 44.7%.

The combined effect of power and heat generation efficiencies is represented in Figure 4.14. The cogeneration efficiencies followed the trend of net effect. Since both the power and heat generation efficiencies increased with increasing turbine inlet temperatures, the cogeneration efficiencies also increased with higher turbine inlet temperatures. Like power thermal efficiency, the heat generation efficiencies were proportional to the turbine inlet temperatures. This analysis showed that in the BCGS, the cogeneration efficiency increased from 79.5% to over 90% and the exergetic cogeneration efficiency increased from 59.8% to 78% at fixed pressure ratio 8.



(a) (b)

Figure 4.14 Effect of gas turbine inlet temperature on cogeneration efficiency in GTCS; (r_p =8)

In the STIG, as Figure 4.14 shows the cogeneration efficiency decreased with the steam injection. This is due to the fact that, in the STIG when the heat demand ratio decreased, both the heat generation energy efficiency and exergetic heat generation efficiency dropped. When the heat demand ratio decreased, due to the steam injection, the turbine exhaust temperature decreased. It is also seen that, the energy and exergy efficiencies are function of the turbine exhaust temperature (T_4). At a fixed turbine inlet temperature pressure ratio (1273 K, 8) and heat demand ratio 0.6, the energy efficiency and the exergetic heat generation efficiency were found to decrease. As a result, the combined effect of power and heat efficiency which is cogeneration energy efficiency decreased from 79.5% to 48.9% whereas exergetic cogeneration efficiency reduced from 59.8% to 46.3%.

4.4 CO_2 Emissions Analysis

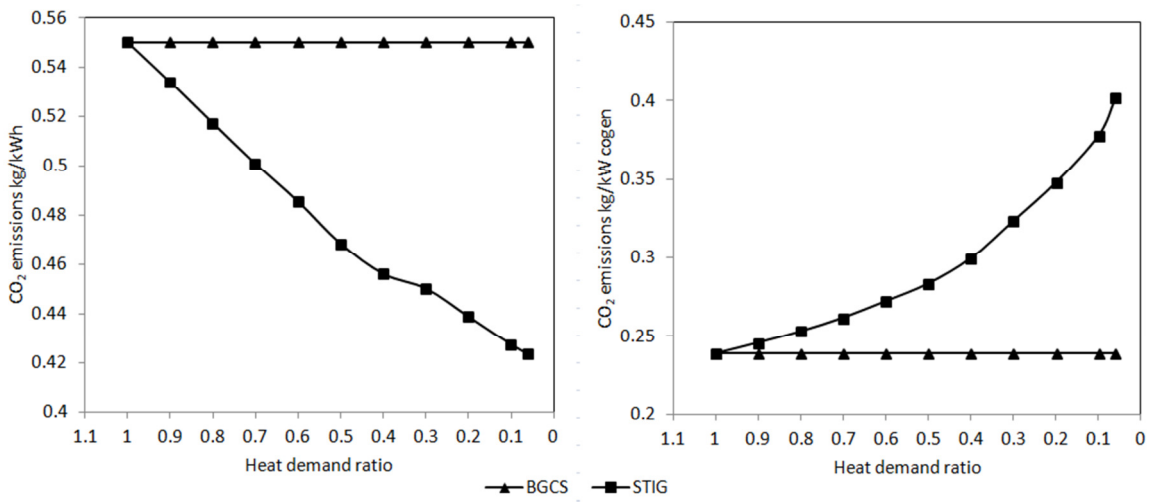
Greenhouse gas emissions are vital to the assessment of future climate change. Carbon dioxide emissions reduction is one of the prime reasons to employ cogeneration approach to power generation. Based on the performance of the BGCS and STIG gas turbine cogeneration systems CO_2 emissions are estimated. The results show that in the STIG gas turbine cogeneration system CO_2 emissions are less than the basic gas turbine cogeneration system. The effect of change in key operating variables on CO_2 emissions was investigated. The emission results are compared for basic and STIG gas turbine cogeneration system. The effect of change in heat demand ratio, pressure ratio and turbine inlet temperature is examined and plotted.

It is important to note that CO_2 emissions are inversely proportional to the efficiencies.
In this analysis the results of the BGCS and STIG gas turbine cogeneration systems are compared with power as well as cogeneration point of view.

4.4.1 Effect of heat demand ratio on CO_2 emissions

Figure 4.15 (a) represents the trend of the emissions per kilowatt-hour power and emissions per kilowatt due to cogeneration. It is seen that in the basic gas turbine cogeneration system CO_2 emissions remains constant even though heat demand ratio is changed. In the basic gas turbine cogeneration system, as the heat demand ratio reduced, the fuel consumption and then power reduced proportionately. So the ratio of power per unit mass of fuel remains constant as a result, and CO_2 emissions 0.55 kg/kWh remained constant.

In the STIG, however as the heat demand ratio decreased, the excess steam is utilized for power augmentation. So the ratio of generated power per kg of fuel increased as heat demand ratio decreased as a result and CO_2 emissions reduced from 0.55 kg/kWh to 0.44 kg/kWh for lowest heat demand ratio of 0.06.



(a) (b)

Figure 4.15. Effect of heat demand ratio on CO_2 emissions in GTCS (rp= 8; TIT=1273K)

Figure 4.15 (b) represents CO_2 emissions in kg per kW cogeneration output. In this figure the combined effect of heat and power output is depicted. In the basic gas turbine cogeneration system the CO_2 emissions 0.239 kg per kW cogeneration remains constant.

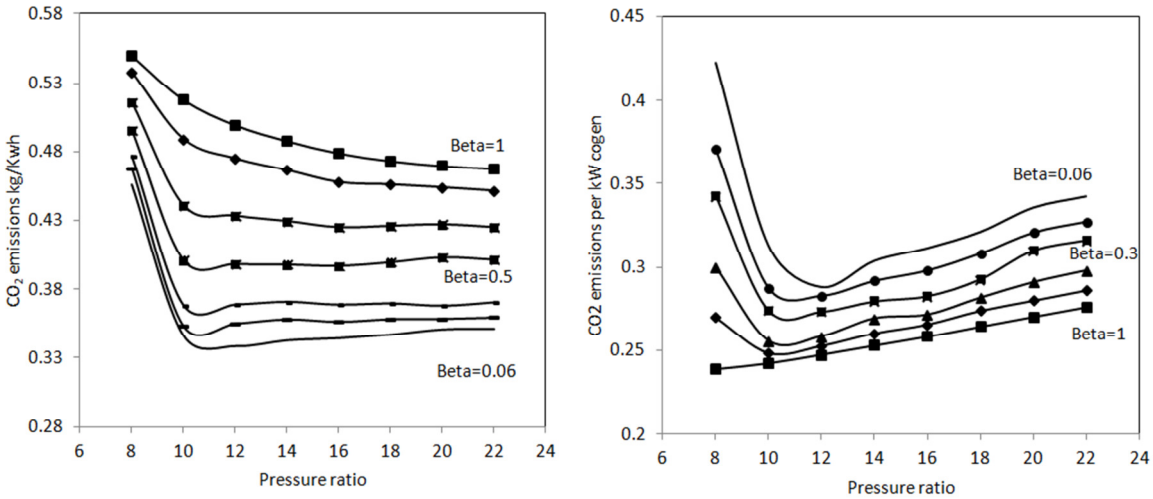
This is due to the fact that as the heat demand ratio decreased, the fuel consumption reduced as a result power and heat output reduced proportionately.

In the STIG however as the heat demand ratio decreased, the quantity of steam supplied to process application decreased and the excess steam is injected in the combustion chamber to increase power. Figure 4.15 (b) shows net effect of cogeneration on CO_2 emissions. In this analysis the CO_2 emissions increased from 0.238 kg/kW to 0.40 kg/kW as the heat demand ratio decreased from 1 to 0.06. Though the trend shows increase in CO_2 emissions, total increased emissions are less compared to CO_2 emissions kg per kWh for power generation only. Thus it is demonstrated that as the heat demand ratio decreases the CO_2 emissions per kW reduced from 0.55 kg/kWh to 0.423 kg/kWh.

4.4.2 Effect of pressure ratio on CO_2 emissions

Figure 4.16 (a) represents the effect of pressure ratio of the gas turbine on CO_2 emissions.It shows that as the pressure ratio increases CO_2 emissions decreased. This is due to the fact that as the pressre ratio incresed the outlet temperature of the compressor increased which reduced the fuel required in the combustion chamber in order to achieve turbine inlet temperature. Thus thermal efficiency increased. Due to reduced fuel consumption CO_2 emissions reduced . In this analysis the pressure ratio varied from 8 to 22. For the basic gas turbine cogeneration system, CO_2 emissions decreased from 0.55 kg/kWh to 0.47 kg/kWh with increase in pressure ratio.

Furthermore in the STIG, it can also be seen that for a given pressure ratio, CO_2 emissions decreases with the heat demand ratio. For a fixed pressure ratio 8 and turbine inlet temperature 1273K, CO_2 emissions was found to decrease from 0.55 kg/kWh to 0.46 kg/kWh. This is because as the heat demand ratio decreases the ratio of generated power to fuel consumption increased,hence CO_2 emissions per kWh decreased.



(a) (b)

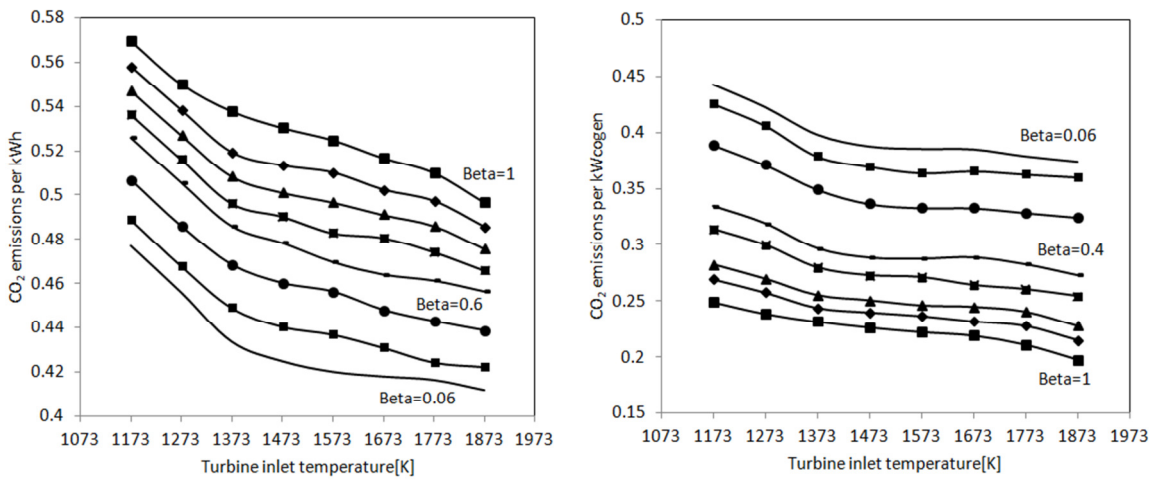
Figure 4.16. Effect of pressure ratio on the CO_2 emissions in GTCS (rp= 8; TIT=1273K)

Figure 4.176 (b) shows the combined effect of pressure ratio on CO_2 emissions considering combined out put of heat and power. It is seen that beyound the optimum pressure ratio the the trend is increasing as the pressure ratio increased. The emission rate per kW cogneration increased due to reduced heat generation efficiency. The heat generation efficiency decreases due to lower turbine exhaust temperatures.The net effect of heat and power output is reduction in cogeneration efficiency. Hence CO_2 emissions have increasing trend for cogeneration. As it is shown in Figure 4.17 (a) for the basic gas turbine cogeneration system, that is when beta is equal to one, CO_2 emissions per kW cogeneration increased from 0.238 kg/kW to 0.276 kg/kW. In the analysis it is demonstrated that, in STIG for a given pressure ratio as heat demand ratio decreased, CO_2 emissions found to be increased. For instance at pressure ratio 8 and TIT 1273K, CO_2 emissions doubled from 0.238 kg/kW to 0.422 kg/kW cogeneration.

4.4.3 Effect of turbine inlet temperature on CO_2 emissions

Effect of turbine inlet temperature on CO_2 emissions is also assessed in this study. Figure 4.17 (a), shows that, the higher turbine inlet temperatures in GT system, lowered the

CO_2 emissions. It can be seen that as the turbine inlet temprature incrased the specific work output kW per kg of gas and thermal efficiency incrased. This implies that more power is generated with consuming same quantity of fuel. Hence CO_2 emissions per kWh decreases. In this analysis, when beta is one that is for the basic gas turbine cogeneration system, CO_2 emissions found to be decreased from 0.57 kg/kWh to 0.49 kg/kWh with increase in the turbine inlet temperature at pressure ratio 8 and TIT 1273K. Further in case of the STIG it is seen that, for a given turbine inlet temperature CO_2 emissions decreased from 0.57 kg/kW to 0.47 kg/kWh with decreasing heat demand ratio.



(a) (b)

Figure 4.17 Effect of TIT on CO_2 Emissions in GTCS (r_p = 8; TIT= 1273K)

Figure 4.17 (b) shows the effect of turbine inlet temperature on CO_2 emissions considoring the combined effect of heat and power. In the BGCS the CO_2 emissions decreases with increase in turbine inlet temperature. It can be seen that when turbine inlet temperature increases the outlet temperature of the turbine incrases so more heat energy is available at HRSG to convert to steam. This increased steam generation at HRSG increases process heat output and heat generation efficiency. Hence CO_2

emissions was found to decrease from 0.57 kg/kWh to 0.47 kg/kWh with increase in turbine inlet temperature from 1173K to 1873K.

In case of the STIG, the CO_2 emissions increased with decrease in heat demand ratio from 0.249 kg/kW to 0.442 kg/kW cogeneration for a fixed turbine inlet temperature and the heat demand ratio (0.06). This is due to the fact that when the heat demand ratio decreased, excess steam is injected into the combustion chamber to augment the power. The output found to be increased without addition of the fuel. This results in reduction in CO_2 emissions.

4.5 Effect of Specific Heat Variation on Performance Characteristics of Gas Turbine Cogeneration Systems

The performance calculations of cogeneration systems based on constant specific heat of air as in air standard cycle often introduces considerable error in estimating performance parameter [37].This is due to the fact that difference in specific heat value between cold air and hot gases. The specific heat of air is independent of gas turbine pressure up to certain operating conditions. However, the specific heat of air greatly influenced by temperature. For instance the specific heat of air at ambient temperature (300 K) is 1.006 kJ /kg K but at 1000K it rose to 1.140 kJ /kg K; increase of about 13.4%. The result shows that the difference between specific heat of air at ambient temperature and specific heat of the combustion products at turbine inlet temperature for instance 1273 K is (1.266 -1.006 =0.26) 26%. This includes specific heats of CO_2 and H_2O which have high specific heat than pure air.

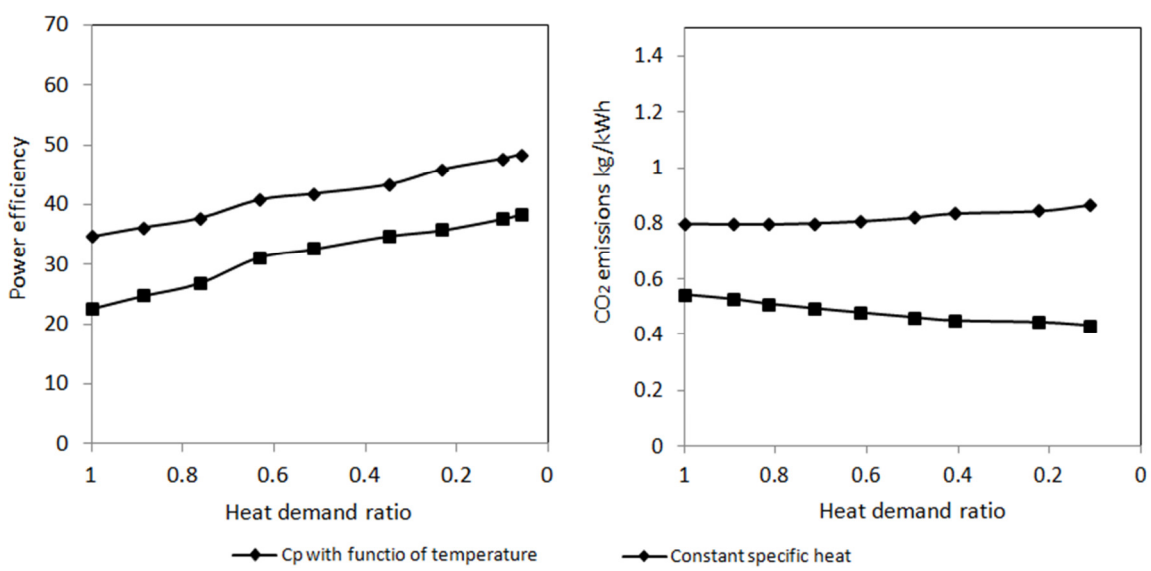
In the present work, both the systems basic and STIG gas turbine cogeneration systems are analyzed. The following two cases are considered:

1. Specific heat as the function of temperature and,
2. with constant specific heat assumption and as that of air.

The results are compared and comparison was plotted in the figures.

4.5.1 Effect of specific heat variation on heat demand ratio, power efficiency and CO_2 emissions

Figure 4.18 shows the impact of specific heat on the performance characteristics for various heat demand ratios. Figure 4.18 (a) represents the relationship between heat demand ratio and power efficiency. The results shows that power efficiency calculated with reference to constant specific heat is much lesser (about 13%) than power efficiency calculated by considering specific heat as function of temperature. Figure 4.18 (b) shows the relationship between heat demand ratio and CO_2 emissions. With constant specific heat the estimated CO_2 emissions were found to be 0.80 kg/kWh, whereas estimated CO_2 emissions considering variable specific heat as function of temperature was 0.55 kg/kWh. The difference in the results got wider as heat demand ratio dropped due to the amount of injected steam in the combustion chamber.



(a) (b)

Figure 4.18 Effect of specific heat on the performance results with regards to heat demand variation in GTCS (r_p =8; TIT = 1273 K)

As Figure 4.18 (b) indicates when heat demand ratio decreased from 1 to 0.06 the results considering specific heat as function of temperature, the CO_2 emissions

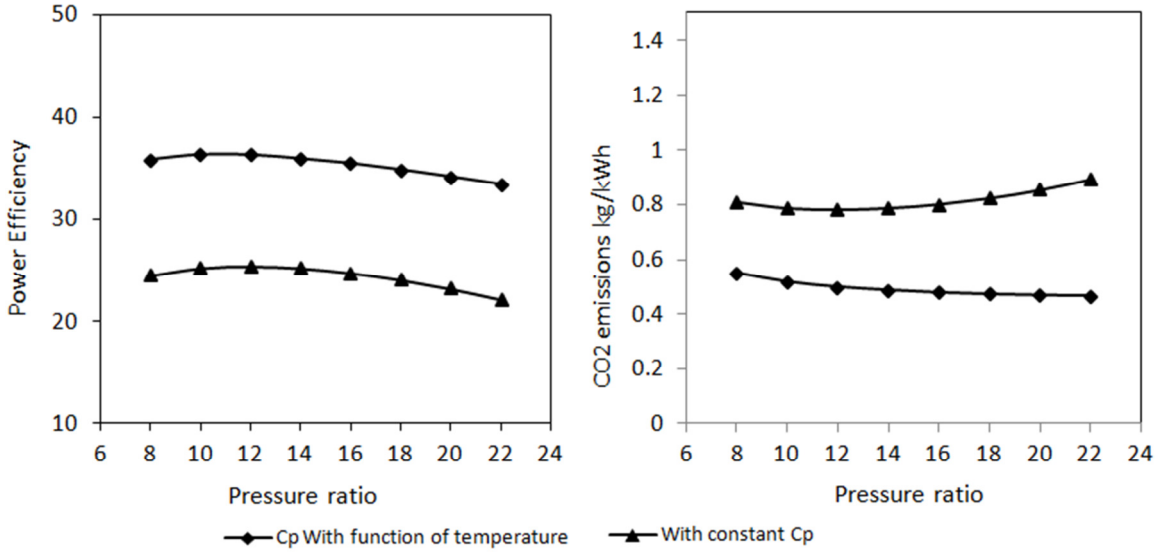
decreased from 0.55 kg/kWh to 0.40 kg/kWh whereas, the results found with constant specific heat consideration are much higher and the trend found to be in the reverse direction. The results show that the CO_2 emissions with constant specific heat consideration were found to increase from 0.75kg/kWh to 0.9kg/kWh.

4.5.2 Effect of specific heat variation on pressure ratio, power efficiency and CO_2 emissions

Figure 4.18 shows the impact of specific heat on the performance characteristics of various pressure ratios. The specific heat variation due to pressure ratio was found to be minimal between 1.235kJ/kg K to 1.266 kJ/ kg K; at TIT 1273 K. Figure 4.19 (a) reveals the impact of specific heat on power efficiency. At 1273K, the efficiency estimated in case of constant specific heat as that of air is lesser than the power efficiency estimated considering the specific heat as the function of temperature. The variation of about 12 % was found between the results.

In case of CO_2 emissions, estimated emissions were found to be in the range of 0.6 to 0.4 kg/kWh considering the specific heat as the function of temperature, while considering constant specific heat, the CO_2 emissions estimated were in the range of 0.7 to 0.9 kg and kWh.

For CO_2 emissions, estimated emissions was found to be in the range of 0.55 to 0.46 kg/kWh considering the specific heat as the function of temperature, while considering constant specific heat, the CO_2 emissions estimated in the range of 0.80 to 0.89 kg and kWh. It is also important to note that, as the pressure ratio increased CO_2 emissions found to be reduced whereas considering with constant specific heat as that of air the results was found to be misleading.



(a) (b)

Figure 4.18 Effect of specific heat on the performance results with regards to pressure ratio variation in GTCS (TIT=1273K)

The specific heat of combustible gases plays an important role in the performance calculations. The results show that there are significant differences in the performance assessment value. The values derived from a constant specific heat assumption do not give correct estimation of the performance. The specific heat calculated based on chemical composition and temperature variation leads to the estimated actual performance.

4.5.3 Effect of specific heat variation on turbine inlet temperature, power efficiency and CO_2 emissions

Figure 4.20 show the impact of specific heat on the performance characteristics of various turbine inlet temperatures. Since the specific heat is a function of temperature, the specific heat variation due to turbine inlet temperatures was found to be significant between 1.235kJ/kg K to 1.396kJ/kg K, at pressure ratio 8.

Figure 4.20 (a) indicates variation in the power efficiency at different turbine inlet temperatures. Considering the specific heat as a function of temperature the power efficiency was found to be in the range of 34.8 % to 39.8 % whereas with constant specific heat considerations the power efficiency was found to be 22.6 to 29.6 % which is much lesser.

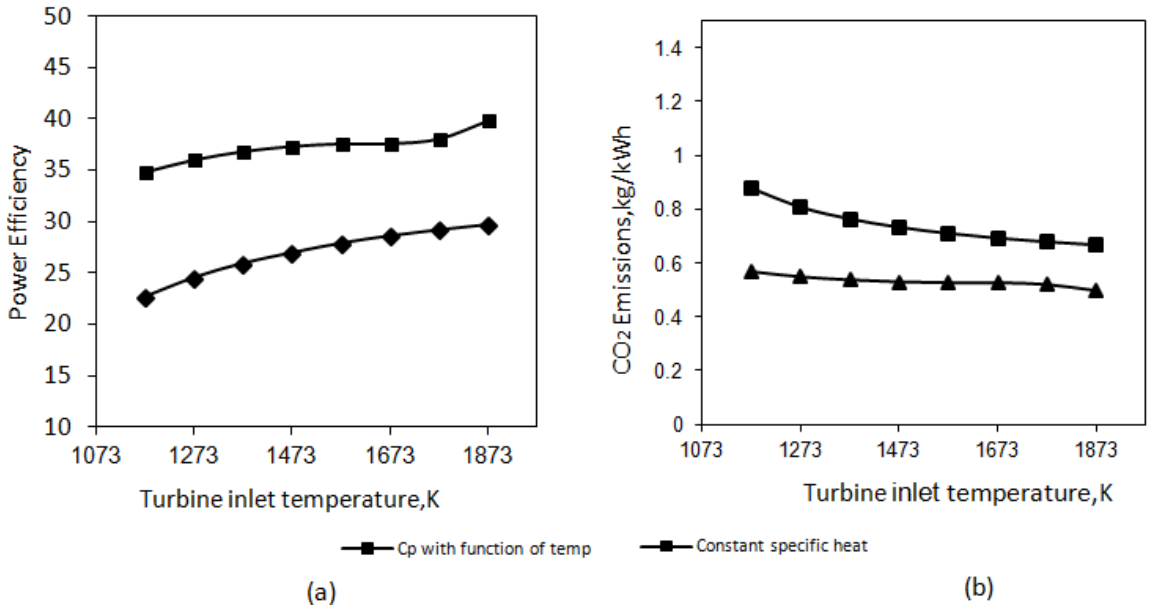


Figure 4.19 Effect of specific heat on the performance results with regards to TIT variation in GTCS (r_p =8; TIT = 1273 K)

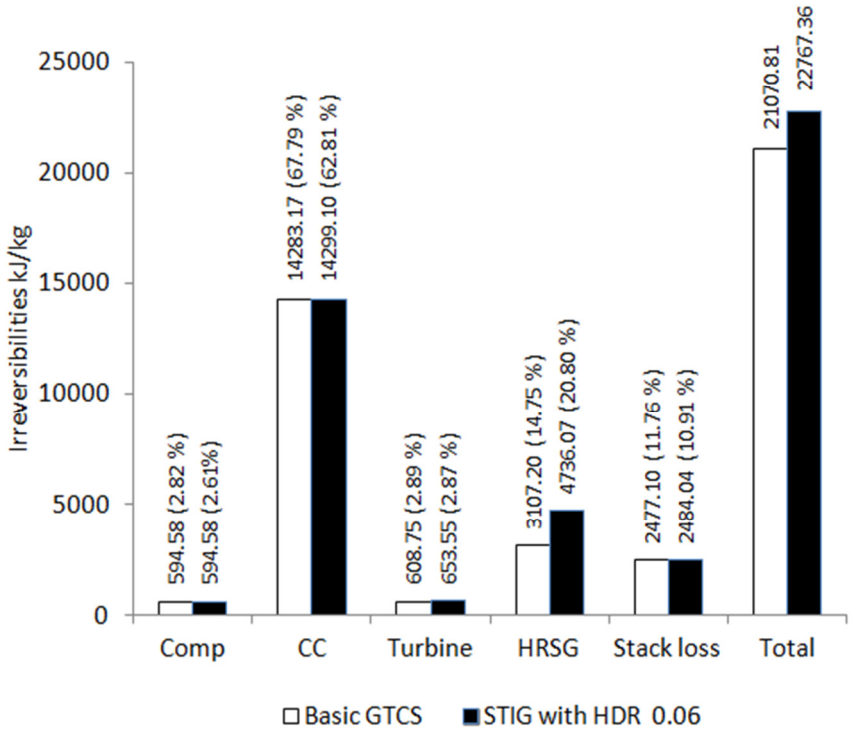
Figure 4.20 (b) shows the comparison between estimated results of CO_2 emissions with specific heat as the function of temperature and constant specific heat at various turbine inlet temperatures. The result shows that CO_2 emissions estimated with specific heat as function of temperature is much less than that of with constant specific heat.

4.6 Exergy Destruction Distribution in Gas Turbine Cogeneration Systems

The exergy analysis of both cogeneration systems was also conducted. In the exergy analysis the exergy destruction rate of individual component in the system was calculated. It is observed that in the both systems the exergy destruction rate of compressor and turbines is as low as 3%. The highest exergy loss was found to be in the combustion chamber (62-67%) followed by the HRSG (14- 21%).

Figure 4.21 shows the comparison between the basic gas turbine cogeneration system and the STIG with regards to component based exergy destruction. The exergy destruction is a function of entropy generation. The highest exergy destruction rate in combustion chamber is due to vigorous and complex combustion process. In the combustion process creates great amount of irreversible entropy [2]. The combustion process is complex chemical reaction characterized by chemical species such as water and carbon dioxide and high temperature.

It can be seen in Figure 4.20, that in the HRSG of steam injected gas turbine cogeneration system, the irreversibilities are more than in the HRSG of BGCS because of the increased entropy generation due to mixture of injected steam and combustion products.



Figure,4.21. Exergy destruction distribution of gas turbine cogeneration systems in GTCS (rp= 8, TIT= 1273K)

In the STIG the exergy destruction rate is 6% more than the exergy destruction rate in the BGCS. It can be concluded that the performance of gas turbine cogeneration system can be optimized by reducing the exergy destruction in the combustion chamber and HRSG.

4.7 Validation of the Results

The results of the computer program were compared to an independent methodology and were found to be in good agreement. The results are validated based on the cases and thermodynamic systems available in the literature. The results were validated by comparing with published results in the literature. In some cases, where data are not available in the literature for validation the results were validated by examining the trends with similar case studies conducted previously.

The balance between thermal and electrical energy supply is key to optimize the performance of cogeneration system, therefore, it is one of the most important criteria in selection of any cogeneration system. The operational characteristics of cogeneration system were examined by varying the heat demand ratio. The heat demand ratio is the ratio of heat demand to the maximum heat that can be supplied for process (heating and cooling application). As it is discussed in Section 4.1; the basic gas turbine cogeneration system was found to be more sensitive to the heat demand variation compared to the STIG. In the case of the STIG, the power output increased with decrease in the heat demand ratio at the same time the estimated drop in heat output is less than the basic gas turbine cogeneration system.

Section 4.2and 4.3 addresses the influence of the pressure ratio and turbine inlet temperature on various performance characteristics of the cogeneration systems such as specific work, energy and exergy efficiencies, and CO_2 emissions. The specific work output increased with increase in pressure up to an optimum pressure ratio then it decreased for higher pressure ratios. The optimum pressure ratio increased in the STIG, as the heat demand ratio decreased. The turbine inlet temperature has positive impact

on the performance of both the BGCS and STIG cogeneration system. Specific work, energy and exergy efficiencies have upward trends with increase in the turbine inlet temperature. These results are in good agreement with the results reported by Kim [27].

In the exergy analysis, in both BGCS and STIG it is observed that the among all the components of the gas turbine cogeneration systems, maximum exergy destruction takes place in combustion chamber, as explained in Section 4.4. This is in agreement with the results published by Bilgen and Kaygusuz [15]. In their work, the performance of the gas turbine cogeneration system based on exergy concept was discussed. It is shown that the combustion chamber accounts for 65% of total exergy destruction in the GT cogeneration system. This was also observed in the study published by Kumar et al. [20].

The CO_2 emission analysis shows that, the emissions were found to be 0.55 kg/kWh at pressure ratio of 8 and TIT of 1273K. The emission count matches with the CO_2 emissions factor published by US Energy information Agency report. In the STIG, CO_2 emissions per unit kWh power reduce with reduction in heat demand ratio, compared to the basic gas turbine cogeneration system. This trend agrees with the work published by Manshoori [29]. In her study, the steam injection system was investigated. The carbon dioxide emissions per kW of the steam injected gas turbine cogeneration system are lower than the gas turbine cogeneration system without steam injection systems.

Significant variations were found between the results achieved by performing calculation on assumption of constant specific heat as that of air and those by considering variation of specific heat of the working media as a function of temperature. The performance results estimated based on constant specific heat gives the misleading picture of the performance. This is validated by performing hand calculations of the specific heat at every state of the working media. The work output from the gas turbine systems is based on the mass flow rate and specific heat of the working fluid. The

specific heat of the methane combustion products is more than the specific heat of air combustion products.

CHAPTER 5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Summary of Results and Conclusions

In the present work the effect of key operating variables such as the heat demand ratio, pressure ratio and turbine inlet temperature on performance characteristics such as specific work, energy and exergy efficiencies and CO_2 emissions of the BGCS and STIG cogeneration systems were assessed. The impact of specific heat variation consideration was also investigated. These conclusions are based on the results obtained by considering the operating conditions discussed in table 3.1.

Heat demand ratio is an important parameter to gauge the flexibility of a cogeneration system in terms of meeting variation in heat and power demand. The basic gas turbine cogeneration system is more sensitive to the power and heat demand as compare to steam injected gas turbine cogeneration system. It is seen that when heat demand ratio decreases the performance of basic gas turbine cogeneration system decreases sharply. In steam injection cogeneration system, when heat demand ratio decreases, the power output increases due to steam injection. The exergetic power efficiency increases due to the steam injection however total exergetic efficiency decreases slightly. The steam injected gas turbine cogeneration system generates less CO2 emissions per kWh.

The turbine inlet temperature is another key operating variable that affects the performance of the cogeneration system significantly. Turbine inlet temperatures have positive influence on specific work, energy and exergy efficiencies. It is also observed that for higher turbine inlet temperature the CO_2 emissions are decreased.

Pressure ratio of gas turbine has a major influence on the performance of cogeneration systems. It is found that, at the optimum pressure ratio, the specific power output increases with increase in steam injection. In the BCGS the maximum specific work found to be 378.63kW/kg. In this investigation, the optimum pressure ratio in the BGCS

was found to be in the range of 10-12. It is also noted that in the STIG the specific work output increased from 378.6kW/kg to 393.9kW/kg as compared to BGCS.

The power and exergetic power efficiency of both cogeneration systems are commensurate with the pressure ratio. It was also observed that, in gas turbine cogeneration systems, for higher pressure ratios, the CO2 emissions are reduced.

In exergy analysis it is found that, in both the gas turbine cogeneration systems the combustion chamber has highest irreversibility. In steam injected gas turbine cogeneration system irreversibility per MWh found less as compared to basic gas turbine cogeneration system.

It is important to consider the specific heat as, function of temperature, to estimate, accurate performance of gas turbine cogeneration systems, the assumption of constant specific heat provide misleading results.

Cogeneration continues to play an important role in controlling the commercial and industrial energy costs through effective system integration of power generation options. It is therefore imperative for power generating companies to generate electricity with high efficiency and in a cost effective manner. In this study it is revealed that the BGCS systems are advantageous in the applications where high and consistent heat demand is required. The performance of the BCGS falls significantly with lower heat demand. One of the options is to go for steam injection. Steam injected gas turbine systems are beneficial thermodynamically, economically and are environmentally friendly too. It is also proved to be reliable and cost efficient.

5.2 Contributions

In comparison with other relevant literature, in his study unique combination of performance parameters are chosen to assess the performance of gas turbine cogeneration systems. To the best of the author's knowledge, energy and exergy analysis of steam injected cogeneration system that takes varying heat demand ratio, and CO_2 emissions into considerations has not been studied. This study shows how both gas turbine cogeneration systems performed under varying heat and power demand and its impact on CO_2 emissions. In addition, in this thesis the influence of operating variables and their effect on CO_2 emissions is discussed which has not been addressed in the open literature.

To my knowledge, in the past nothing much has been written about the importance of specific heat consideration. In this study the influence of specific heat considerations is addressed. This study shows that the importance of considering the specific heat as a function of temperature in a more accurate estimation of the performance results.

5.3 Recommendations for Future Work

In this thermodynamic analysis calculations were performed based on simulated conditions. In order to simplify analysis, the state conditions were defined and the appropriate assumptions made. Due to the time and resources constraints it was not possible to analyze real cogeneration systems. This work can be further extended by analyzing actual gas turbine cogeneration systems taking actual state conditions and heat losses into account. The performance results of actual cogeneration systems will provide useful data for

a. Optimizing the operating parameters for a cogeneration system under consideration, and provide the estimated performance data prior to making costlier capital investment.
b. Those who would like to retrofit existing basic cogeneration systems.

In this work the comparison between the basic gas turbine cogeneration system and steam injected gas turbine cogeneration system was performed. A wider range of cogeneration systems based on variety of fuels or prime mover should be considered. The comparison will be useful in appropriate cogeneration system selection, based on CO_2 emission reduction, with regards to the Kyoto Protocol on the climate change.

The exergy analysis in this thesis work can be further applied to the existing cogeneration systems in order to tap the components with high irreversibilities in the cogeneration systems and taking measures to reduce them.

Lastly, the design and analysis of various thermal systems can be studied further with the help of the approach and methodology used in this analysis.

REFERENCES

1. International Panel for Climate Change forth assessment report, climate change, causes of change, 2007.
2. International energy outlook 2013, DOE/EIA 0484, 2013.
3. Taylor, P., Energy efficiency indicators for public electricity production from fossil fuels, IEA information paper, 2008
4. Eurelectric "Preservation of Resources" Working Group's "Upstream" Sub-Group in collaboration with VGB, A report: Efficiency in electricity generation, July 2003.
5. Moran, M. and Shapiro, H. Fundamental of Engineering Thermodynamics, 5th Edition, Wiley, Danvers, MA, 2004.
6. Catalog of CHP Technologies, US Environmental Protection Agency, Combined Heat and Power Partnership, December 2008.
7. Canadian Industrial Energy, End-Use Data and Analysis Center. A review, of existing cogeneration facilities in Canada, March 2011.
8. Nag P., Power Plant Engineering, 3rd Edition, Tata McGraw-Hill, New Delhi, 2007.
9. Wina Graus, Ernst Worell, Comparison of efficiency, fossil fuel power generation, Ecofys, Utrecht, 2006.
10. Cogeneration: A Primer, Canadian Gas association, 2013.

11. Brooks, F., GE gas turbine characteristics, GE power systems, GER 3567H, October 2000.
12. Pak, P. S., and Suzuki, Y., Exergetic evaluation of gas turbine cogeneration systems for district heating and cooling, Int. Journal of Energy Res., Vol. 21, 209-220, 1997.
13. Reddy, B. V., and Butcher, B., Second Law analysis of a natural gas-fired gas turbine cogeneration system, Int. Journal of Energy Res., Vol.33, 728-736, 2009.
14. Bilgen, E., Exergetic and engineering analyses of gas turbine based cogeneration systems, Energy, Vol. 25, 1215-1229, 2000.
15. Khaliq, A., and Kaushik, S., Thermodynamic performance evaluation of combustion gas turbine system with re-heat, Applied thermal engineering, Vol.24, 1785-1795, 2004.
16. Khaliq, A., and Choudhary, K., Combined first and second law analysis of gas turbine cogeneration system with inlet air cooling and evaporative after cooling of the compressed discharge, Journal of Eng. gas turbine power, Vol. 129, 1004-1011, 2007.
17. Reddy, B.V., Ramkiran, Kumar, A, Nag, P., K., Second law analysis of a waste heat recovery steam generator, Int. Journal of mass and heat transfer, Vol. 45, 1807-1814, 2002.

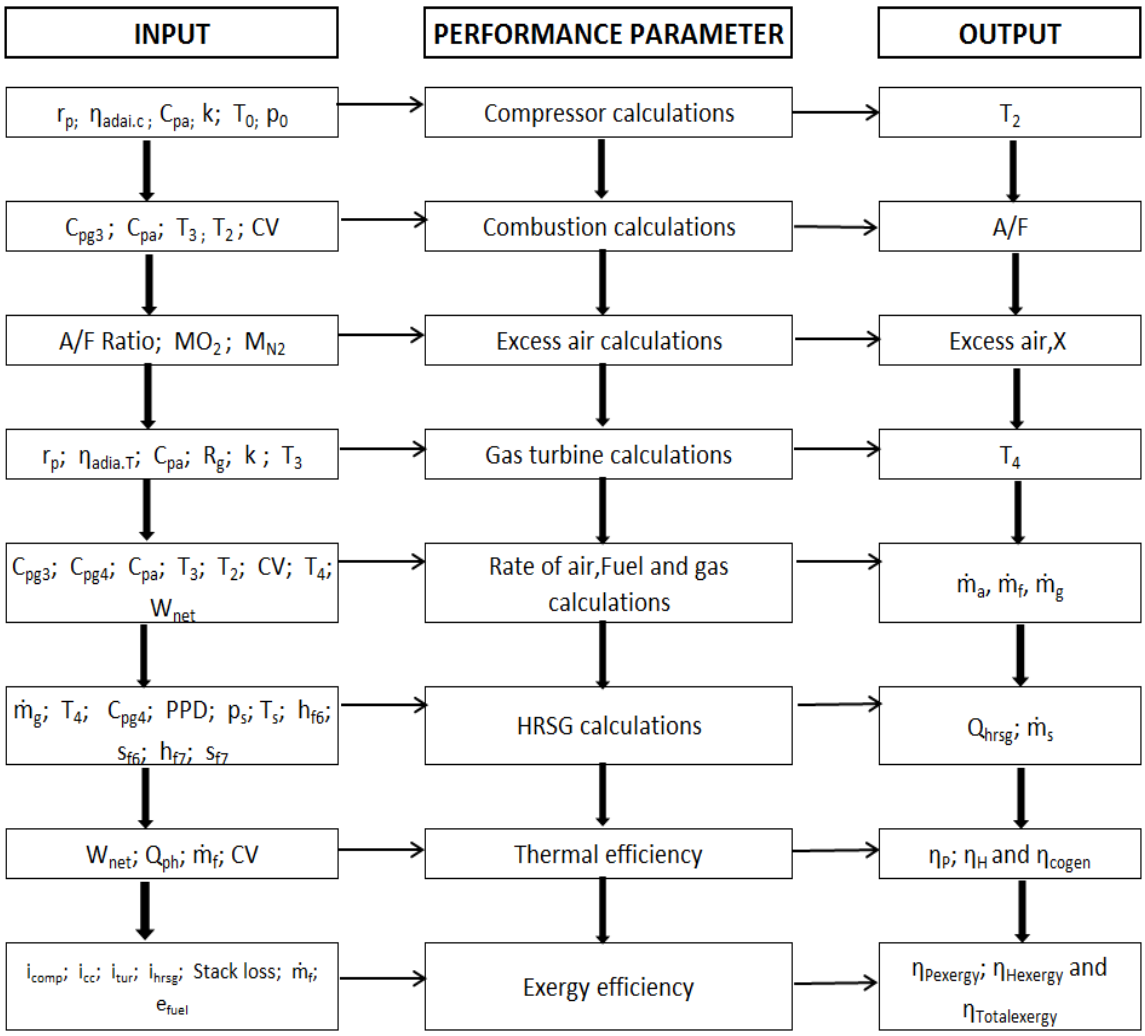
18. Rahman, M., Ebrahim, T. and Abdella, A. Thermodynamic performance analysis of gas turbine power plant, International sciences of physical sciences, Vol. 6, 3359-3550, 2011.
19. Ganapati, V., Options for improving the efficiency of the Heat Recovery Steam Generators, Electric energy publications Inc., 2011.
20. Kumar, A., Kachhwaha, S. and Mishra, R. Thermodynamic analysis of a regenerative gas turbine cogeneration plant, Journal of scientific and Industrial Research, Vol. 69, 225-231, 2010.
21. Srinivas, T., Gupta, A. and Reddy, B.V., Parametric simulation of steam injected gas turbine combined cycle, Proceedings of the institution of mechanical engineers , Part A, Journal of power and energy , Vol.221, 873-883, 2007.
22. Rice, I., G., Steam injected gas turbine analysis: Steam rates, Journal of Engineering for gas turbine and power, ASME, Vol.117, 347-353, 1995.
23. Soufi, M., Fuji, T., and Sugimoto, K., A modern injected steam gas turbine cogeneration system based on exergy concept, International Journal of energy research, Vol. 28,1127-1144, 2004.
24. Jonson, M., and Yan, J., Humidified gas turbines – a review of proposed and implemented cycles, Energy, Vol. 30, 1013-1078, 2005.
25. Aissani, S., Bouam, and Kadi, R., Evaluation of gas turbine performances and NO_x and CO emissions during steam injection in the upstream of combustion chamber. Syrian renewable energy conference, 2010.

26. Kim, K., and Giman, K., Thermodynamic performance assessment of steam injection gas turbine Systems, World academy of science, engineering and technology. Vol. 4, 945-951, 2010.
27. Wang, F., and; Chiou, J., Performance Improvement for a simple gas turbine GENSET-a retrofitting example, Applied Thermal Engineering, Vol.22, 1105-1115,2002.
28. Manshoori, N., Tafazoli, D., and Gazikhani, M., Experimental Investigation of steam injection in GE F5, Gas Turbine NOX Reduction Applying Vodoley System, Second International Conference on Applied Thermodynamics, Istanbul, 2005.
29. Motahar, S., and Alemrajabi, A., Performance augmentation of an aero engine gas turbine using steam injection, International conference of Energy and Environment, 2006.
30. Agarwal, S., and Mishra, R., Thermodynamic analysis for improvement in thermal performance of simple Gas turbine cycle through retrofitting techniques, Proceedings of the National conference on trends and advances in Mechanical engineering, YMCA University of science and technology, Faridabad, Haryana, 2012.
31. Nishida, K., Takagi, T., and Kinoshita, S., Regenerative steam injection gas turbine systems, Applied Energy, Vol.81, 231-246, 2005.

32. Rosen, M., Energy and environmental advantages of cogeneration over Nuclear and coal Electrical utilities, Proceedings of the 4th IASME / WSEAS international conference of energy and environment, 2009.
33. Dincer, I., Energy and environmental impacts: present and future perspectives, Energy sources, Vol.20, 427-453, 1998.
34. Dincer, I., Thermodynamics, Exergy and environmental impact, Energy sources, Vol. 22,723-732, 2000.
35. Coleseus, C., and Shepherd, S., The Cheng cycle offers flexible cogeneration options, International power technology, reprinted from modern power systems, 1985.
36. Energy and environmental analysis, Technology, Characterization: Micro turbines, Environmental protection agency, combined heat and power partnership program, 2008.
37. Tara chand, V., Ravi Sankar, B., and Rangaraya chowdary, J., Exergy Analysis of Gas Turbine Power Plant, International Journal of engineering trends and technology, Vol.4, 3991-3993, 2013.
38. Atouei, S., Fasihfar, A., Malektash, S., Energy and exergoeconomic analysis and optimization of steam injected gas turbine cogeneration systems using genetic algorithm, Journal of power and electrical engineering, Vol. 1, 1-9, 2014.

Appendix Flow Charts

A.1 Basic Gas Turbine Cogeneration System Simulation Flow Chart



A.2 Steam Injection Gas Turbine Cogeneration System Simulation Flow Chart