ME Dissertation

Entitled

THERMODYNAMIC ANALYSIS OF GAS TURBINE COGENERATION SYSTEM INTEGRATED WITH IAC AND STIG

By

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In
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CERTIFICATE

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Date: ___________

This is to certify that the dissertation entitled "**THERMODYNAMIC ANALYSIS OF GAS TURBINE COGENERATION SYSTEM INTEGRATED WITH IAC AND STIG**" submitted by Mr. LALIT MALI, Roll No. 12296, in partial fulfillment for the award of the degree of Master of Engineering in Thermal Engineering, is an authentic record of student's own work carried out by him under our guidance and supervision. He has completed his work with utmost sincerity and diligence.

It is also certified that this dissertation has not been submitted to any other Institute/University for the award of any degree or diploma.

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CANDIDATE’S DECLARATION

I hereby declare that the work which is being presented in this project report entitled “**THERMODYNAMIC ANALYSIS OF GAS TURBINE COGENERATION SYSTEM INTEGRATED WITH IAC AND STIG**” submitted as M.E. dissertation towards the fulfillment of the requirements for the award of the degree of Master of Engineering with specialization in Thermal Engineering, D.C.E. Delhi, is an authentic record of my own work carried out under the supervision of **Dr. S.S. Kachhwaha**, Assistant Professor and **Dr. R.S. Mishra**, Professor in the Department of Mechanical engineering, Delhi college of engineering, Mechanical Engineering Department, Delhi.

The matter embodied in this dissertation report has not been submitted by me for the award of any other degree or diploma.

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It is a great pleasure to have the opportunity to express my heartiest felt gratitude to everybody who helped me throughout the course of this dissertation.

The feeling of acknowledgement and its expression in words are entirely different. It is a matter of profound privilege and pleasure to express my deep sense of gratitude and indebtedness to my guides **Dr. S.S. Kachhwaha**, Assistant Professor and **Dr. R.S. Mishra**, Professor in the Department of Mechanical engineering, Delhi college of engineering, for there invaluable guidance, encouragement and patient review. Their continuous inspiration only has made me complete this project.

I would also like to take this opportunity to present my sincere regards to all my teachers for their kind support and encouragement.

Last, but not the least, I would like to mention that all the blessing and affection of my parents, my colleagues and my seniors encouraged me to always work hard.

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ABSTRACT

With a high back-work ratio and a high exhaust temperature, the simple cycle gas-turbine generation system usually has a low generation-efficiency especially when the ambient temperature is hot. The exhaust gases of a gas turbine carry a significant amount of thermal energy that is usually expelled to the atmosphere without taking any further part in the power generation process. The low grade thermal energy can however be put to beneficial use. To recover the very high temperature of exhaust gases (around 500˚C) from simple cycle gas turbine generation sets (GENSETs), a heat recovery steam generator (HRSG) is often used and steam is generated. The generated steam can be either used for many useful processes (heating, drying, separation etc.) or used back in the power generation system for enhancing power generation capacity and efficiency.

Among many technologies to improve the efficiency of a simple cycle gas turbine, inlet air cooling (IAC) and steam injection gas turbine (STIG) are considered the best ways to modify an existing simple cycle unit without major destruction to its original integrity. These attractive features can use the steam generated in the HRSG to improve the power generation capacity and efficiency.

The objective of the present work is to explore the utilization of the exhaust gases of a gas turbine plant. In this study, a 7B Frame GENSET was considered as base unit and IAC & STIG features have been sequentially analyzed after system retrofitting with the concept of cogeneration without steam injection, cogeneration with steam injection and cogeneration with IAC & process steam cycle. The present study is to evaluate the system performance parameters (first-law and second-law efficiency, power-to-heat ratio, COP, fuel-air ratio, steam injection ratio, compressor work, turbine work and net power output) in different cases after system retrofitting. For the simulation of power generation system, a computer program has been developed in Engineering Equation Solver (EES) software and validated with the data available in the literature.

Parametric study has been performed using combined first and second law approach to investigate the effects of compressor air inlet temperature and pressure of steam on first and second law efficiency, power-to-heat ratio, fuel-air ratio, steam injection ratio and exergy destruction in the system components.

Keywords: Cogeneration, Gas turbine, Heat recovery, IAC, STIG, Exergy Destruction

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NOMENCLATURE

AP	approach point (˚C)
C_p	specific heat of the fluid (kJ/kg)
E	exergy (kJ)
$\dot{E}$	exergy rate (kW)
e	specific chemical exergy (kJ/kmol)
f_c	circulation rate
h	specific enthalpy of the fluid (kJ/kg)
l	fraction of heat loss
M	molecular weight (kg/kmol)
$\dot{m}$	mass flow rate (kg/s)
n	molar flow rate (kmol/s)
P	pressure (bar)
PP	pinch point (˚C)
Q	heat transfer (kJ)
$\dot{Q}$	heat transfer rate (kW)
$\bar{R}$	universal gas constant (kJ/kmolK)
r_p	pressure ratio of the compressor
s	specific entropy of the fluid (kJ/kgK)
T	temperature (K)
TIT	Turbine Inlet Temperature (K)
U	internal energy (kJ)
v	specific volume (m^3/kg)
W	work (kW)
$\dot{W}$	work transfer rate
w	steam injection ratio
x	molar fraction
X	concentration of LiBr
y	exergy destruction ratio

Subscripts

1-24 & f	state points in Fig. 4.1
a	air
ABS	Absorber
C	Compressor
CC	Combustion Chamber
COND	Condenser
con	condensate
cv	control volume
D	destruction
ECO	Economizer
el	electrical
EVAP	Evaporator
f	fuel
g	Electric Generator
GEN	Heat Generator
HRSG	Heat Recovery Steam Generator
k	kth component
L	loss
manf	manufacturer
o	reference
p	product or gases
P	pinch point
pro	process
r	refrigerant
REG	Regenerator
S	isentropic
SHX	Solution Heat Exchanger
sat	saturated
sup	superheated
s	steam
T	turbine
Th	thermal
w	water

Superscript

’	fraction of gas phase at dead state
1	fraction of gas phase before combustion
2	fraction of gas phase after combustion
CH	chemical
KN	kinetic
PH	physical
PT	potential

Greek symbol

η	efficiency
λ	fuel-air ratio
ε	effectiveness of the regenerator

Acronyms and abbreviations

AHE	Air Heat Exchanger
ARS	Absorption Refrigeration System
BLR	Boiler
CCPT	Combined Cycle Power Plant
CHP	Combined Heat and Power
COP	Coefficient of Performance
EES	Engineering Equation Solver
GENSET	Generation Set
GT	Gas Turbine
HHV	Higher Heating Value
IAC	Inlet Air Cooling
IEA	Indian Electricity Authority
LHV	Lower Heating Value
MVC	Mechanical Vapour Compression
RSTIG	Regenerative Steam Injected Gas Turbine
SPHR	Superheater
STIG	Steam Injected Gas Turbine
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el	electrical
EVAP	Evaporator
f	fuel
g	Electric Generator
GEN	Heat Generator
HRSG	Heat Recovery Steam Generator
k	k^{th} component
L	loss
manf	manufacturer
o	reference
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CHAPTER 1

INTRODUCTION

1.1 Cogeneration- Definition and Historical Development

Cogeneration is an engineering concept involving the production of both thermal energy, generally in the form of steam or hot water, and electricity in one operation, thereby utilizing fuel more effectively than if the desired products were produced separately. The ratio of electric power to thermal energy varies depending on the plant type. A cogeneration plant may be conceived to supply thermal energy or electric power. In the first case, electric power is considered to be a by product and is relatively small. In the latter case, which is often encountered by public utility companies, thermal energy is considered as a by product. There are conceptually different cogeneration plants: the steam turbine based, gas turbine based, and diesel engine based plant. The gas turbines differ in power output, cycle efficiency, cycle pressure ratio, firing temperature, exhaust temperature and exhaust flow rate.

Cogeneration first appeared in late 1880's in Europe and U.S.A. During the early parts of the 20th century, most industrial plants generated their own electricity using coal-fired boilers and steam turbine generators. Many of the plants used the exhaust steam for industrial processes. In 1903, a Norwegian, Egidius Elling, was able to build the first gas turbine that was able to produce more power than needed to run its own components, which was considered an achievement in a time when knowledge about aerodynamics was limited. Using rotary compressors and turbines, it produced 11 hp (massive for those days). His work was later used by Sir Frank Whittle.

It has been estimated that as much as 58% of the total power produced by on-site industrial power plants in the U.S.A. in the early 1900's was cogenerated. In addition to decreased fuel consumption, cogeneration results in a decrease of pollutant emissions. For these reasons, governments in Europe, U.S.A. and Japan are taking an active role in the increased use of cogeneration. Research, development and demonstration projects realised during the last 25 years led to a significant improvement of the technology, which is now mature and reliable. New techniques are also under development, such as fuel cells.

demonstration projects realised during the last 25 years led to a significant improvement of the technology, which is now mature and reliable. New techniques are also under development, such as fuel cells.

The era around 1940 is known to be golden for developing gas turbine cogeneration plant. As, in this time, the oil and gas has a prime and profitable market. Refineries need lots of process heat as well as power for its operation. Thus in refineries, cement plants and chemical industries the gas turbine cogeneration plant have been used frequently by their designer. Now a day's, the gas turbine cogeneration plants are being developed to a new horizon as the issues are now changing towards gaining more efficiency, reduction of fuel cost and protection of environment from the combustion gases.

Industrial growth in India and the change in living styles of people since independence, i.e. in the last four decades, have resulted in a steady rise in the demand for power. Consequently, the power sector has registered a phenomenal growth. Despite this growth in power generation, there is always a gap in the demand for power and the power generated. At present this shortfall is estimated to be in the range of about 12000 MW. India is the second largest producer of sugar in the world and there exists a huge potential of generating electricity to the extent of about 15000 MW through cogeneration in core industries such as sugar and rice mills, distilleries, petrochemical sector and industries such as fertilizers, steel, chemical, cement, pulp and paper, and aluminum.

An attempt has been made to implement a suitable cogeneration scheme in a sugar factory to generate power for not only meeting the requirements of the sugar factory but also feeding about 4000 kW of excess power to the state electricity grid. This project is drawing great attention from the government's financial institutions, technologists and engineers from all over the country. CHP currently accounts for around 9 % of global power generation (IEA 2007). Its economic potential, however, is likely to be significantly greater.

1.2 Gas Turbine Cogeneration Systems Integrated with IAC and STIG

Man has a natural tendency to try to improve the efficiency of devices which convert thermal energy into mechanical work. Moreover the rise in fossil fuel costs is a stimulus for this kind of research. The energy consumption of the world has been increased very rapidly. It appears that an energy gap, i.e., the difference between consumption and the naturally growing production of non-renewable sources of energy might develop. In order to bridge this gap, and also consider the additional problems of atmospheric pollution, some policy should be considered.

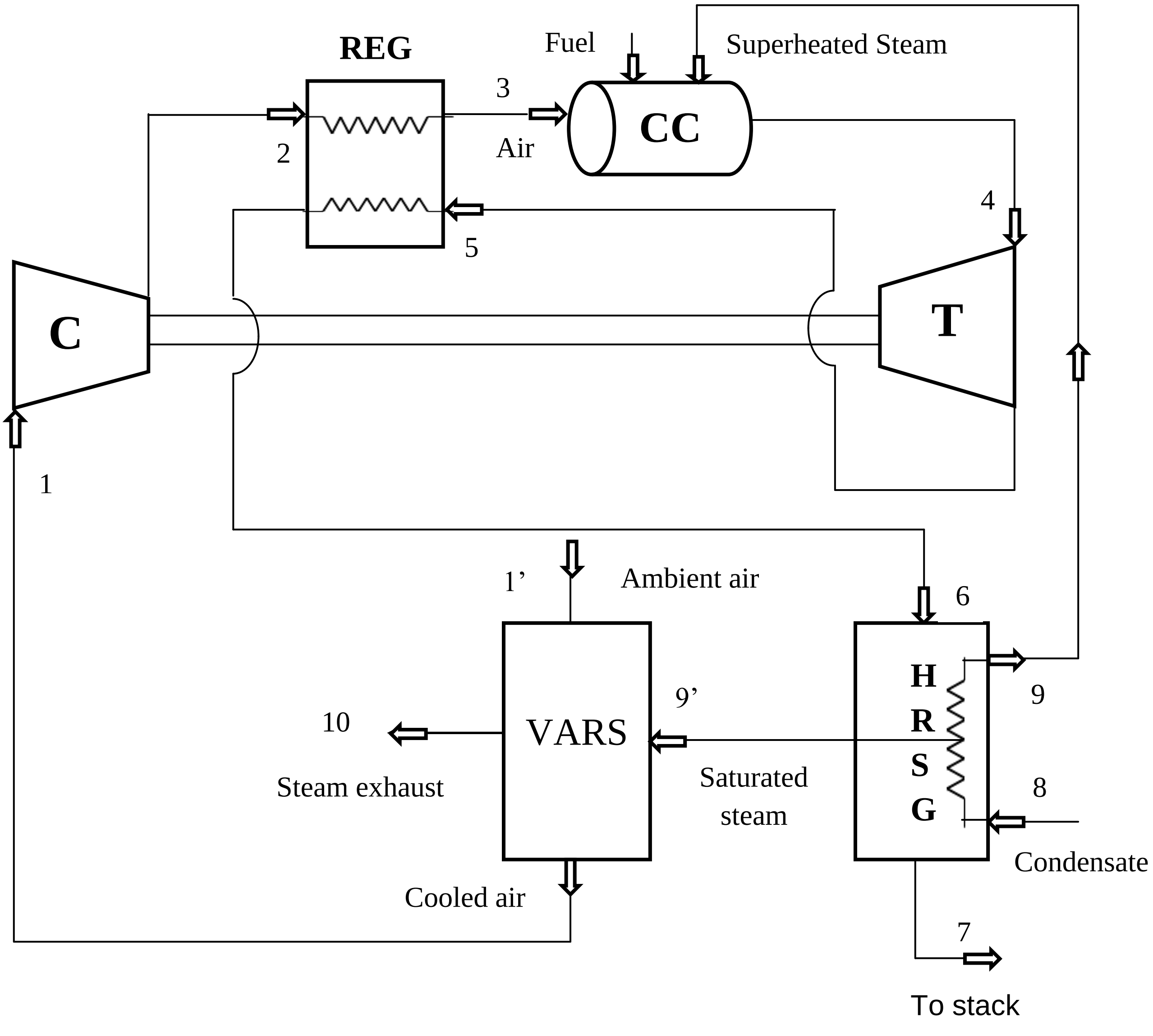

Fig.1.1 Elementary Diagram of Cogeneration System with Inlet Air Cooling and Steam Injection

Many simple cycle gas turbine generation sets are used to serve as peak load units which can be started quickly but suffer from very low efficiency, especially during summer peaking hours when electricity is most needed. To mitigate the anticipated power shortage, retrofitting projects have been seriously considered to convert theses existing simple cycle gas turbines into more advanced cycle units with higher efficiency and higher power output.

The thermodynamic processes of a simple cycle gas turbine can be approximately modelled as a Brayton cycle, in which the back work ratio is usually very high, and exhaust temperature is often above 500˚C. A high exhaust temperature implies that there is plenty of useful energy wasted to the environment. The recovery of this otherwise wasted energy can be used to improve either the power generation capacity or/and efficiency via modification to the basic cycle, such as gas to gas recuperation, steam injection, evaporation cycle, chemical recuperation, inlet air cooling and combined cycle.

Among many other technologies, the IAC and STIG also have become common practices to enhance the performance of power generation. Both features are very cost effective and can be implemented in the basic system without major modifications to the original system integration.The IAC method stands for inlet air cooling. The IAC technology is simple to cool down the air entering the compressor. With a cooler inlet air, the compressor consumes less work and can compress more air per cycle to increase the capacity of gas turbine.

Different options including evaporative cooling, mechanical chiller, absorption chiller, vapour absorption refrigeration system and thermal energy storage in gas turbine power augmentation using IAC are available. Among these, recovering the waste heat to generate steam that in turn, powers a vapour absorption refrigeration system is naturally matched with the use of STIG. Use the same HRSG to recover most of useful energy and generate as much steam as possible, the high pressure steam being injected into the combustor and the lower pressure steam being used to power the vapour absorption refrigeration system.

The STIG method stands for steam injected gas turbine. The steam generated from heat recovery steam generator (HRSG) is injected into the combustion chamber. Air from the compressor and steam from the HRSG both receive fuel energy in the combustion chamber and both expand inside the same turbine to boost the power output of turbine. The STIG method is a very effective way to boost the net power output and increases the overall efficiency of gas turbines. In fact, STIG has become a well established practice.

Although many efforts have been devoted to either apply the STIG technology or the IAC method to enhance gas turbine performance, little has ever integrated STIG and IAC for the same system. Since the energy levels required by IAC and STIG are different, the recovered energy could be more fully utilized by a combined STIG and IAC features.

1.3 Features and Applications of Gas Turbine

Gas turbines have been developed as heavy duty units either for industrial and utility applications, or as lightweight, compact and efficient aircraft engines. These engines are modified for stationary applications, in which case they are called aero derivative turbines. In general, they are capable of faster start-ups and rapid responses to changing load, therefore suitable to be used as a load-follow unit. Gas turbine designs have been successfully used for cogeneration having main advantages like low initial cost, compact, high availability, fast and low-cost maintenance, fuel-switching capabilities, high quality heat which can be easily recovered, and high efficiencies. In addition, the commercial availability of packaged units helped in their widespread applications. Unfortunately, the simple cycle GENSET has low generation efficiency, especially when the ambient weather is hot (the time when electrical power is most needed). Therefore, gas-turbine GENSETs are mostly on standby unless there is a power-shortage problem or a black out emergency.

The characteristics of gas turbines which has been retrofitted with the concept of steam injection and inlet air cooling are given below-

1. Characteristics of the GE MS5002 gas turbine, ref **[15]**

P_{manf}	18,000 kW
$\eta_{Th\text{-}manf}$	23%
r_p	7.38
TIT	900˚C
η_C	90%

- η_{CC}	95%
- η_T	88%

2. Features of the Allison 501 KH, at the maximum steam injection, ref **[12]**

- P_{el}	5330 kW
- P_C	5490 kW
- $\dot{m}_s$	2.349 kg/s
- r_p	11.7
- TOT	494˚C

3. Features of the 7B GENSET, ref **[11]**

- TIT	1264 K
- r_p	7.38
- η_C	90%
- η_T	88%
- P_s	1.41 MPa

1.4 Motivation of the Present Research Work

Gas turbine cogeneration system is used extensively now a day's all around the world. Due to integration of various applications, there is growing demand for power as well as process heat, for various industries like oil refining, power plant, cement plant, mining, refrigeration etc. Power can be produced with the help of gas turbines, as they are easy to install, quick starting, cheaper and smaller. However, the cost and starting time is an advantage for the gas turbines to use as a power-producing device but there are some setbacks in gas turbines. Major setback in case of gas turbines is the compressor which takes approximately two third of turbine power to drive itself, which contributes to major loss. Other one is the high temperature of exhaust gases (around 400-600˚C) leaving through the turbine. The heat of waste exhaust flue gases can be further utilized as a process heat. Sometimes, low-pressure steam is also generated with the help of waste heat recovery boiler. Further, this steam can be used as a heating medium in various applications or converted to superheated steam and used to drive a steam turbine; this configuration is known as combined power cycle in which both steam and gas turbines are contributes in generating power. Though high temperature gas heat is utilized to form the steam or in process heating but even the exhaust gas temperature from stack has to be maintained high enough (above 130˚C), so that dew formation of any gas can be avoided which further ensures the long life of the stack.

Various research and development work is going on around the world to reduce the compressor work and simultaneously reduce the heat supplied. The performance of gas turbines can be enhanced by doing minor modifications in gas turbines without major destruction to its original integrity. One of the most interesting aspect of gas turbine is that waste heat of flue gases can be used in generating steam at two different pressures. Higher pressure steam can be used for injection in combustion chamber and low pressure steam can be used for running the vapour absorption refrigeration system. These attractive features can improve the power generation capacity and efficiency to a considerable extent. Again, steam injection can reduce NO_x emissions. These enhancements are nowadays in application in some parts of the world and expected to widen throughout the world because gas turbine is thermo economically advantageous. This is the main motivation for doing this research work.

1.5 Organization of the Report

Chapter 1 expresses the introduction of cogeneration, its history and development, review of gas turbine cogeneration system integrated with STIG and IAC and its importance. A brief explanation about features and applications of gas turbine, motivation of present research work, various manufactures data regarding gas turbines in which steam injection and inlet air cooling has been implemented are given.

Chapter 2 shows literature review section in which various research related to steam injection and inlet air cooling simple gas turbine plant. Also the conclusion of the study has been expressed in this chapter.

Chapter 3 describes formulation of cogeneration system in which standard cogeneration system integrated with both IAC and STIG features has been discussed. Also the assumptions made in mathematical formulation, detailed explanation of formulation of system components has been carried out.

Chapter 4 covers program validation part for cogeneration system with results and discussions. Also case study on cogeneration with the concept of inlet air cooling and steam injection has been studied. In the first phase, computer program algorithm has been developed for the system. In the second phase, program validation was carried out by supplying various input parameters to the program. The results for various parameters have been tabulated and comparison was made between computer program results and results of reference at state points of the system.

Chapter 5 describes the general conclusions and recommendations for future work of present study.

CHAPTER 2

LITERATURE REVIEW

Gas turbine cogeneration is an extensive area of research, in which several researchers have already made some mark and several others are indulging themselves in this direction. Literature survey for the project is enlisted below.

2.1 Summary of Literature Review

Major area of research in gas turbine cogeneration included the research on cogeneration plant and refrigeration system to improve the performance thermodynamically with reduced expenditure.

Bejan et al **[1]** established a design methodology for the gas turbine cogeneration system. This simple system is integrated with regenerator and HRSG is attached to utilize the waste heat. A 30 MW gas turbine is designed on the basis of analysis of enthalpy and entropy of air and gas. Simultaneously, the exergy (physical and chemical) and exergy destruction at different points are also tabulated. The waste heat from turbine (exhaust gases) has been used to produce steam for other processes.

Sioh et al **[2]** worked out on exergy analysis of gas turbine cogeneration system. The paper examined performance of gas turbine cogeneration systems well as the exergy destruction in each component in the system when it is operated at part and full load conditions. In addition, the effect of inlet air temperature, humidity of the inlet air, water and steam injection on the performance of the system is analyzed. The predicted values of the performance for 1MW gas turbine cogeneration system have been compared with the actual performance data provided by the gas turbine manufacturer. Computerized energy and exergy analysis in his work provides efficiency calculation, exergy destruction in each component, mass flow rate, composition and exhaust temperature.

Bilgen **[3]** performed exergetic and engineering analyses of gas turbine based cogeneration systems. This paper presents exergetic and engineering analyses as well as a simulation of gas turbine-based cogeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine. The exergy analysis is based on the first and second laws of thermodynamics. The engineering analysis is based on both the methodology of levelized cost and the payback period. To simulate these systems, an algorithm has been developed.

Alexis **[4]** studied performance parameters for the design of a combined refrigeration and electrical power cogeneration system. This paper discusses the conservation of energy in a cogeneration system. A steam power cycle (Rankine) produces electrical power 2 MW and steam is bleeded off from the turbine at 7 bar to warm a factory or units of building during the winter or to supply a steam ejector refrigeration cycle to air-conditioning the same area during the summer. In the summer this system can be as alternative solution instead of absorption. A computer program has been developed for the stufy of performance parameters of the cogeneration system.

Khaliq and Kaushik **[5]** studied the thermodynamic performance evaluation of combustion gas turbine cogeneration system with reheat. The paper has presented a methodology based on first and second law for the thermodynamic performance evaluation, of combustion gas turbine cogeneration system with reheat. The energetic and exergetic efficiencies have been defined. The effects of process steam pressure and pinch point temperature used in the design of heat recovery steam generator, and reheat on energetic and exergetic efficiencies have been investigated.

Wang et al **[6]** studied the economic feasibility of waste heat to power conversion. In this paper, individual effect of IAC and STIG after system modification was evaluated by the simulation of the power generation system and the ABSIM code developed by Oak Ridge National Laboratory was adopted to simulate the absorption refrigeration system. In this study, an existing simple cycle GE MS7001B generation-system was considered as the basic system and converted into the modified system with either IAC or/and STIG features.

Kakaras **[7]** studied compressor intake-air cooling in gas turbine plants. The purpose of this work is to present a computer simulation of the integration of an innovative technology for reducing the intake-air temperature in gas turbine plants. Simulation results for two test cases are presented: a simple cycle gas turbine and a combined cycle plant. First, the effect of ambient air temperature variation on the power output and efficiency is presented for both cases. Next, the results from the integration of an evaporative cooler and air-cooling system under consideration are presented and discussed, demonstrating the gain in power output and efficiency that can be achieved.

Lucia et al **[8]** discussed the benefits of compressor inlet air cooling for gas turbine cogeneration plants. This paper presents a comparative analysis of different solutions for cooling the compressor inlet air for the LM6000 gas turbine in a cogeneration plant operated in base load. Absorption and evaporative cooling systems are considered and their performance and economic benefits compared for the dry low-NOx LM6000 version. Reference is made to two sites in Northern and Southern Italy. The results confirmed the advantages of inlet air cooling systems.

Ondryas et al **[9]** investigated gas turbine power augmentation in a cogeneration plant using inlet air chilling. Options include absorption chillers, mechanical (electric driven) chillers, thermal energy storage. Motive steam for the chillers is steam from the gas turbine exhaust or electrical energy for mechanical chillers. Chilled water distribution in the inlet air system is described. The overall economics of the power augmentation benefits is investigated. Results from the analysis revealed that the gas turbine power augmentation via inlet air chilling can be effectively used to boost power during high ambient temperature.

Khaliq **[10]** performed the exergy analysis of gas turbine trigeneration system for combined production of power heat and refrigeration. In this paper, a conceptual trigeneration system is proposed based on the conventional gas turbine cycle for the high temperature heat addition while adopting the heat recovery steam generator for process heat and vapour absorption refrigeration for cold production. Combined first and second law approach is applied and computational analysis is performed to investigate the effects of overall pressure ratio, turbine inlet temperature, pressure drop in combustor and heat recovery steam generator, and evaporator temperature on the exergy destruction in each component, first law efficiency, electrical to thermal energy ratio, and second law efficiency of the system.

Wang and Chiou [**11]** conducted the integration of steam injection and inlet air cooling for a gas turbine generation system. In this study, an existing frame 7B simple cycle GENSET was considered as the basic system and converted into the modified system with either the IAC or/and STIG features. The steam needed in the STIG and IAC features is generated from the energy recovered from the system's own exhaust gases. Under the conditions of local summer weather, the benefits obtained from the system implementing both STIG and IAC features are more than a 70% boost in power and 20.4% improvement in heat rate.

Salvi and Pierpaoli **[12]** presented the optimization of inlet air cooling systems for steam injected gas turbines. The cooling of the inlet air in gas turbines is a practice used to improve power performance. For the purpose of verifying the positive effects of compression air cooling in STIG turbines, a calculation model has been developed and studied specifically for gas turbines and STIG turbines. The model has been applied to the Alison 501 KH turbine. In addition to focussing on the two traditional techniques used to cool the air of the compression system through an absorption unit and an intercooling unit, the present work emphasizes the advantages of using an ejection cooling system.

Nishida et al **[13]** analyzed the regenerative steam-injection gas-turbine systems. In this paper, the performance characteristics of two types of regenerative steam-injection gas-turbine (RSTIG) systems are analyzed and the thermal efficiencies and specific powers were investigated. Furthermore, the performances of these systems are compared with those of the simple, regenerative, water injection and steam injected gas-turbines (STIG) cycles.

Srinivas et al **[14]** performed sensitivity analysis of STIG based combined cycle with dual pressure HRSG. Steam from high-pressure steam turbine is injected into the combustion chamber at a pressure higher than the combustion pressure to improve the exergy efficiency of combined cycle. The effect of steam injection mass ratio, deaerator pressure (or temperature ratio), steam reheat pressure ratio, HP steam turbine pressure, compressor pressure ratio and combustion temperature on combined cycle exergy efficiency has been investigated.

Bouam et al **[15]** performed combustion chamber steam injection for gas turbine performance improvement during high ambient temperature operations. The main purpose of the present work is to improve the principal characteristics of gas turbine used under stringent condition of temperature in Algerian Sahara by injecting steam in the combustion chamber. The suggested method has been studied and compared to a simple cycle. Efficiency, however, is held constant when the ambient temperature increases from ISO conditions (101.3 kPa, 288K, 60% RH) to 50°C. Computer program has been developed for various gas turbine processes including the effects of ambient temperature, pressure ratio, injection parameters, standard temperature, and combustion chamber temperature with and without steam injection.

Khaliq and Choudhary **[16]** worked out on combined first and second-law analysis of gas turbine cogeneration system with inlet air cooling and evaporative aftercooling of the compressor discharge. In this paper, a conceptual gas turbine based cogeneration cycle with compressor inlet air cooling and evaporative aftercooling of the compressor discharge was proposed to increase the cycle performance significantly and render it practically insensitive to seasonal temperature fluctuations. Computational analysis is performed to investigate the effects of the overall pressure ratio, turbine inlet temperature and ambient relative humidity on the exergy destruction in each component, first law efficiency, power-to-heat ratio, and second law efficiency of the cycle using combined first and second-law approach.

Mostafavi et al **[17]** performed the thermodynamic analysis of combined open-cycle-twin-shaft gas turbine (Brayton cycle) and exhaust gas operated absorption refrigeration unit. This paper explores the utilisation of the exhaust gases of an open-cycle-twin-shaft gas turbine. There is sufficient amount of energy in the exhaust gases for precooling purposes. This energy can be introduced to an absorption unit and by considering the efficiency of heat exchangers and the coefficient of performance of the absorption refrigeration machine, the amount of cooling capacity available in the exhaust gases can be used for charge air cooling or air conditioning or both. An air standard cycle is assumed for the gas turbine, first with the aid of thermodynamic laws the specific network and the efficiency of the cycle as a function of temperature ratio and pressure ratio of the cycle are calculated, and the realistic bounds placed on the cycle by the thermodynamic analysis is shown.

AL-Hawaj et al **[18]** analyzed a cogeneration scheme comprising a combined cycle power plant (CCPT) with an absorption chiller used for space cooling. This paper suggests a new scheme for utilizing GT turbines to meet the peak load demand during summer season while increasing the utilization factor of such plant. A parametric study investing the effects of different parameters, such as steam to gas mass flow rate, fraction of turbine steam extraction, ambient temperature, inlet steam turbine temperature, compressor pressure ratio, and gas turbine (GT) combustion efficiency on the performance of the system has been made. In another aspect of the study, the relative advantage of using CCPP with absorption cooling over thermally equivalent mechanical vapour compression (MVC) cooling is also studied.

Talbi and Agnew **[19]** performed exergy analysis on a single-effect absorption refrigeration cycle with lithium-bromide-water as the working fluid pair, an optimisation procedure that consists of determining the enthalpy, entropy, temperature, mass flow rate, heat rate in each component and coefficient of performance has been performed.

Kachhwaha et al **[20]** worked out on second law analysis of single stage water/lithium bromide vapour absorption refrigeration system. A thermodynamic model to simulate a single stage refrigeration machine of 10 kW capacity with H_2O-LiBr has been developed. The present model evaluates the thermodynamic state (pressure, concentration) of the four main elements of the machine (generator, condenser, evaporator, and absorber) once the relative temperatures are known. Thermodynamic properties of each state point in the system are calculated using related equations of state. Heat transfer rate of each component in the system and some performance parameters (circulation ratio f_c, COP) are calculated from the first law analysis.

Kilic and Kaynakli **[21]** performed second law-based thermodynamic analysis of single stage water/lithium bromide absorption refrigeration system. In this study, the first and second law of thermodynamics are used to analyze the performance of single-stage water-lithium bromide absorption refrigeration system (ARS) when some working parameters are varied. A mathematical model based on the exergy method is introduced to evaluate the system performance, exergy loss of each component and total exergy loss of all the system components. Finally, this study draws attention to a detailed thermodynamic analysis (first and second law) and leads the way to improve thermal systems and their components.

Ait **[22]** studied simple gas turbine system with inlet air refrigeration by vapour compression cycle. In this paper, power boosting of gas turbine are analyzed due to inlet air refrigeration. Mass flow rate term has been replaced by volumetric flow rate, thus reducing the inlet air temperature increases the volumetric flow rate. The results show improvement in terms of cycle efficiency and specific power output.

2.2 Conclusions of Literature Review

The researchers in the field of gas turbine cogeneration system have already delivered a vast variety of work. However, the possibility of improving the domain of work is still there by combining different methods and methodologies for enhancing the performance of the system. The main conclusions from the Literature review are pointed below-

- Exergy analysis of gas turbine cogeneration system has been performed by various researchers based on first and second law of thermodynamics in which exergy destruction in each and every component of the system has been calculated when it is operated at part and full load conditions. Opportunities are available to analyze different gas turbine cogeneration system viz. simple cycle, gas turbine cogeneration system with alternative regeneration, gas turbine cogeneration system with regeneration based on the same input parameters, to analyze the superiority of any particular method in power generation, economy and environmental issues.
- Simulation and dynamic modelling of gas turbine-based cogeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine has been worked out.
- The concept of reheat has been applied in evaluating thermodynamic performance of combustion gas turbine cogeneration system.
- Individual effect of IAC and STIG after system modification has been studied by the simulation of the power generation system. System with STIG features have the best generation efficiency (improved from 29.3% to 39.9 %), while the system with both STIG and IAC can achieve the greatest power capacity (increased from 52.1 MW to 96.8 MW). System with the STIG can further reduce NOx emissions below 25 ppm without the help of extra clean-up equipment. System with both IAC and STIG features can greatly reduce the exergy loss of the stack exhaust from 1.04 MW per MW output.
- Turbine power augmentation has been performed in a cogeneration plant using inlet air chilling. Options include absorption chillers, mechanical (electric

driven) chillers, thermal energy storage. Simulation has been done on simple cycle gas turbine and a combined cycle plant.

- The performance characteristics of regenerative steam-injection gas-turbine (RSTIG) systems have been analyzed. The performances of these systems have been compared with those of the simple, regenerative, water injection and steam injected gas-turbines (STIG) cycles.
- Analysis of gas turbine cogeneration system with inlet air cooling and evaporative aftercooling of the compressor discharge has been performed and studied by combined first and second-law.

2.3 Objectives of the Present Research Work

- To analyze and design gas turbine cogeneration system integrated with IAC and STIG.
- To develop formulation of various components of gas turbine cogeneration system by combined first and second law analysis.
- To develop a computer based design methodology for a gas turbine cogeneration system integrated with IAC and STIG.
- To perform a parametric study for wide range of variables such as compressor air inlet temperature, compressor pressure ratio, turbine inlet temperature, regenerator effectiveness etc. The effect of these parameters on first law efficiency or fuel utilization efficiency, second law efficiency or exergetic efficiency, specific fuel consumption, power to heat ratio etc. has been studied. Exergy destruction in each and every component of gas turbine and vapour absorption refrigeration system has been calculated.

CHAPTER 3

FORMULATION OF COGENERATION SYSTEM

3.1 Cogeneration Systems Description

The standard cogeneration system integrated with both IAC and STIG features is shown in Figure 4.1. The top side of Figure 4.1 is the basic unit, which includes a compressor, a combustor, a gas turbine and a generator. The air at atmospheric condition passes through the evaporator of vapour absorption refrigeration system from where the air gets cool by the refrigerating effect and then it enters to the air compressor from where the air is compressed to high pressure. Air after compression in the compressor enters the regenerator where its temperature is raised by utilizing the energy of the exhaust gases exiting from turbine, and then it enters to the combustion chamber where the fuel is added. Due to combustion of fuel, the temperature of air further increases. Now the mixture of air and gases formed due to combustion of fuel enters the turbine where they expand and produce the work output (alternator to generate electricity). In order to recover the energy from the exhaust gases, a heat-recovery steam generator (HRSG) was installed at the downstream exit of the turbine.

The maximum energy that can be recovered by the HRSG is limited by the effectiveness of the HRSG and the outlet temperature of the flue gas. The effectiveness of the HRSG is set to be 0.8, and the exhaust from the stack is taken as 130˚C to prevent the possibility of vapour condensation, ref **[10]**.

A typical single stage vapour absorption refrigeration system is used to cool the inlet air of the power generation system. The total heat required by the vapour absorption refrigeration system to cool the compressor inlet air is only a small fraction of the available energy. In order to utilize fully the recoverable energy, the STIG mechanism was also implemented in the system. In other words steam at two different pressures will be used during operation.

Higher pressure (at 14 bar) steam (superheated steam) is used for STIG, ref **[5]** which is generated in the HRSG directly by utilizing the energy of exhaust gases. Lower pressure (at 8 bar) steam (saturated steam) is used for running the vapour absorption refrigeration system, ref **[5]** which is generated by utilizing the energy of available superheated steam after deciding the quantity of steam required for steam injection. Ultimately, the exhaust gases from HRSG are expelled to the atmosphere through the stack.

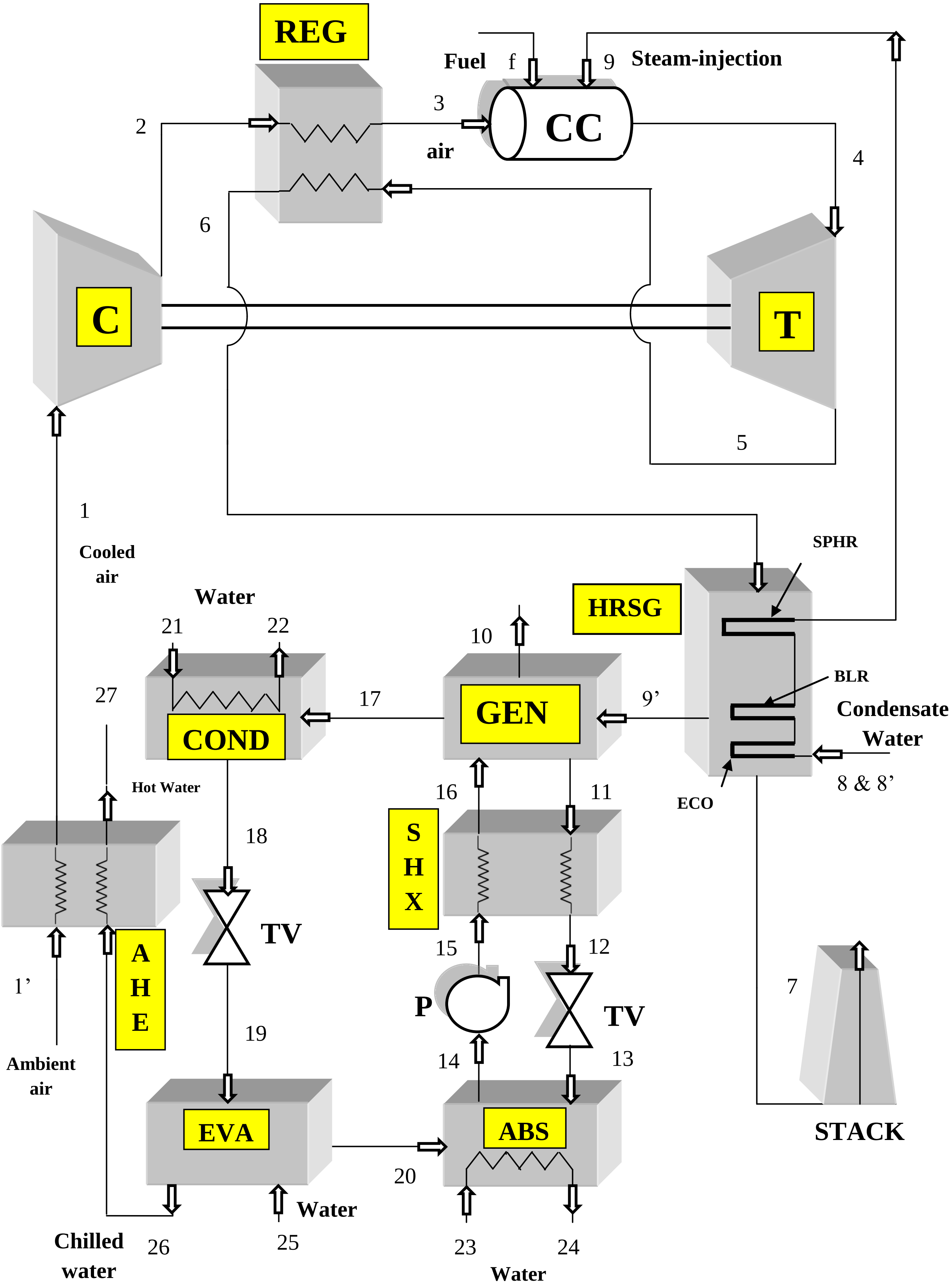

Fig. 3.1 Cogeneration system integrated with IAC and STIG

State points in Fig. 4.1-

1	compressor inlet
1’	air inlet to evaporator
2	compressor outlet or regenerator inlet
3	regenerator outlet or combustion chamber inlet
4	combustion chamber outlet or turbine inlet
f	fuel inlet to combustion chamber
5	turbine outlet
6	HRSG inlet
7	HRSG outlet
8	condensate inlet to HRSG at high pressure
8’	condensate inlet to HRSG at low pressure
9	superheated steam inlet to combustion chamber
9’	saturated steam inlet to generator
10	saturated steam outlet to generator
11	inlet of weak water concentration LiBr solution to solution heat exchanger
12	inlet of weak water concentration LiBr solution to throttle valve
13	outlet of weak water concentration LiBr solution from throttle valve
14	inlet of strong water concentration LiBr solution to pump
15	inlet of strong water concentration LiBr solution to solution heat exchanger
16	inlet of strong water concentration LiBr solution to generator
17	refrigerant inlet to condenser
18	refrigerant outlet from condenser or refrigerant inlet to throttle valve
19	refrigerant outlet from throttle valve or refrigerant inlet to evaporator
20	refrigerant outlet from evaporator
21	cooling water inlet to condenser
22	cooling water outlet from condenser
23	cooling water inlet to absorber
24	cooling water outlet from absorber
25	water inlet to evaporator
26	water outlet from evaporator
27	water outlet from air heat exchanger
1’	air inlet to air heat exchanger

3.2 Mathematical Formulation

The mathematical formulation of the present analysis is based on the following assumptions, ref **[1]**-

1. The cogeneration system operates at steady state.
2. Ideal-gas mixture principles apply for the air and the combustion products.
3. The fuel (natural gas) is taken as methane modelled as an ideal gas. The fuel is provided to the combustion chamber at the required pressure by throttling from high-pressure source.
4. The combustion in the combustion chamber is complete and N_2 is assumed as inert gas.
5. Heat loss from the combustion chamber is 2% of the fuel lower heating value. All other components operate without heat loss.
6. Combustion chamber has been maintained at constant temperature in the presence of steam (in case of steam injection), ref **[14]**.
7. Vapour absorption refrigeration system handles the binary mixture of water and lithium bromide where the water as refrigerant and lithium bromide as absorbent is used.
8. The pressure drop in vapour absorption refrigeration system is neglected.
   The HRSG unit is a single pressure counter current heat exchanger with fixed effectiveness.
9. The HRSG unit is a single pressure counter current heat exchanger with fixed effectiveness.

The thermodynamic analysis of the proposed system has been carried out using equation of mass and energy balance. For this purpose, computer programme has been developed in Engineering Equation Solver (EES) in which the control volume model of each component was constructed using mass, energy and exergy balances for determining the thermodynamic properties at every key position for the system as shown in Figure 3.1.

A set of governing equations for a particular component (k) is expressed as-

Mass rate balance

$$\sum_{k} \dot{\mathrm{m}}_{\mathrm{i,k}} = \sum_{e} \dot{\mathrm{m}}_{\mathrm{e,k}} \tag{3.1}$$

Energy rate balance

$$\dot{Q}_{cv,k} - \dot{W}_{cv,k} = \sum_{e} \dot{m}_{e,k} h_{e,k} + \sum_{i} \dot{m}_{i,k}\, h_{i,k} \tag{3.2}$$

Exergy rate balance

$$\dot{E}_{D,k} = \sum \dot{E}_{q,k} - \dot{W}_{cv,k} + \sum_{i} \dot{E}_{i,k} - \sum_{e} \dot{E}_{e,k} \tag{3.3}$$

Where $\dot{E}_D$ denotes the rate of exergy destruction and $\dot{E}_q$ denotes the associated exergy transfer rate due to heat transfer.

If the effect of kinetic and potential energy is ignored, the total exergy rate $\dot{E}_k$ consisting of physical and chemical can be expressed as

$$\dot{E}_k = \dot{E}_k^{PH} + \dot{E}_k^{CH} \tag{3.4}$$

The specific formulation for various system componenets are discussed below-

3.2.1 Compressor

Air enters the compressor at ambient conditions. This initial temperature and mass of air dictate the amount of work required for compression, the fuel that can be burnt, the fuel required to achieve a specified turbine inlet temperature. As a result, the net power output, the efficiency, the exhaust gas flow rate and temperature at the turbine exit (consequently the recoverable heat) are functions of the ambient conditions.

In a compressor, a gas is caused to flow in the direction of increasing pressure and/or elevation by means of a mechanical or electrical power input which is given by

$$\dot{W}_C = \dot{n}_a(\overline{h}_2 - \overline{h}_1)$$

$$= \dot{m}_a \frac{\overline{h}_2 - \overline{h}_1}{M_a} \qquad (3.5)$$

Where $\dot{n}_a$ is molar flow rate of air, $\dot{m}_a$ is mass flow rate of air, M_a is molecular weight of air and $(\overline{h}_2 - \overline{h}_1)$ is the difference of enthalpy between states 1 and 2. The value of specific enthalpy $\overline{h}_2$ can be evaluated using the compression isentropic efficiency. Solving the expression for compression for compressor isentropic efficiency

$$\eta_{SC} = \frac{\overline{h}_{2S} - \overline{h}_1}{\overline{h}_2 - \overline{h}_1} \qquad (3.6)$$

We get

$$\overline{h}_2 = \overline{h}_1 + \frac{\overline{h}_{2S} - \overline{h}_1}{\eta_{SC}} \qquad (3.7)$$

Where η_{SC} is isentropic efficiency of compressor, $\overline{h}_1$ and $\overline{h}_2$ are enthalpies of air at state 1 and 2. $\overline{h}_{2S}$ is the isentropic enthalpy at compressor exit.

As the air composition is fixed, ideal gas mixture principles allows the isentropic compression to be described as-

Change in entropy during isentropic compression in compressor

$$\overline{s^0{}_{2S}} - \bar{s}_1 = X_{1,N2}\left[\bar{s}^0(T_{2s}) - \bar{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{N_2}$$

$$+ X_{1,O2}\left[\bar{s}^0(T_{2s}) - \bar{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{O2}$$

$$+ X_{1,CO2}\left[\bar{s}^0(T_{2s}) - \bar{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{CO2}$$

$$+ X_{1,H2O}\left[\bar{s}^0(T_{2s}) - \bar{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{H2O}$$

$$= 0 \qquad (3.8)$$

where

$X_{1,N2}$ = mole fraction of N_2 before combustion (= 0.7748)

$X_{1,O2}$ = mole fraction of O_2 before combustion (= 0.2059)

$X_{1,CO2}$ = mole fraction of CO_2 before combustion (= 0.0003)

$X_{1,H2O}$ = mole fraction of H_20 before combustion (= 0.019) and

$\bar{R}$ = universal gas constant (8.314 kJ/kmolK)

Using the inbuilt specific entropy expressions for N_2, O_2, CO_2 and H_2O in software, the above expression is solved for T_{2S} and corresponding enthalpy of $\bar{h}_{2S}$ is determined.

Physical Exergy at state 1-

At this state T_1 = T_O and $P_1 = P_O$. Accordingly, $\bar{h}_O = \bar{h}_1$ and $\bar{s}_1 = \bar{s}_o$ therefore the physical exergy component vanishes.

$$\dot{E}_1^{PH} = 0 \qquad (3.9)$$

Chemical exergy at state 1-

$$\dot{E}_1^{CH} = 0 \qquad (3.10)$$

Total exergy at state 1-

$$\dot{E}_1 = \dot{E}_1^{PH} + \dot{E}_2^{CH} = 0 \qquad (3.11)$$

Physical exergy at state 2-

$$\dot{E}_2^{PH} = \dot{m}_a \frac{\bar{h}_2 - \bar{h}_0 - T_0(\bar{s}_2 - \bar{s}_0)}{M_a} \qquad (3.12)$$

Chemical exergy at state 2-

$$\dot{E}_2^{CH} = 0 \qquad (3.13)$$

Total exergy at state 2 can be similarly calculated as state 1.

Exergy destruction in compressor-

$$\dot{E}_{D,C} = \sum_i \cancel{\left(1 - \frac{T_0}{T_1}\right)} Q_1 - \dot{W}_C + \dot{E}_1 - \dot{E}_2$$

$$\dot{E}_{D,C} = \dot{W}_C + \dot{E}_1 - \dot{E}_2 \qquad (3.14)$$

3.3.2 Regenerator

Regenerator is a type of heat exchangers. The purpose of the heat exchangers is to increase the exergy of the hot stream coming from compressor and entering into the combustion chamber. Preheating the air promotes better combustion.

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} \qquad (3.15)$$

Physical exergy at state 3-

$$\dot{E}_3^{PH} = \dot{m}_a \frac{\overline{h}_3 - \overline{h}_0 - T_0(\overline{s}_3 - \overline{s}_0)}{M_a} \qquad (3.16)$$

Chemical exergy at state 3-

$$\dot{E}_3^{CH} = 0 \qquad (3.17)$$

Total exergy at state 3 can be similarly calculated as state 1.

Exergy destruction in regenerator-

$$\dot{E}_{D,REG} = \sum_i \cancel{\left(1 - \frac{T_0}{T_1}\right)} Q_1 - \cancel{\dot{W}_C} + \dot{E}_2 + \dot{E}_5 - \dot{E}_3 - \dot{E}_6$$

$$\dot{E}_{D,REG} = \dot{E}_2 + \dot{E}_5 - \dot{E}_3 - \dot{E}_6 \qquad (3.18)$$

3.3.3 Combustion Chamber

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

$$\frac{n_f}{n_a} = \lambda \tag{3.19}$$

and $$\frac{n_p}{n_a} = 1 + \lambda \tag{3.20}$$

where the subscripts f, p and a denote fuel, combustion products & air respectively and λ is the fuel-air ratio.

For complete combustion of natural gas (methane) with steam injection in the combustion chamber, chemical equation takes the following form

$$\begin{aligned} &\bar{\lambda} CH_4 + \left[X_{1,N2} N_2 + X_{1,O2} O_2 + X_{1,CO2} CO_2 + X_{1,H2O} H_2 O\right] + wH_2O \\ &\rightarrow \left[1 + \bar{\lambda} + w\right] \quad \left[X_{2',N2} N_2 + X_{2',O2} O_2 + X_{2',CO2} CO_2 + X_{2',H2O} H_2 O\right] \end{aligned} \tag{3.21}$$

Balancing carbon, hydrogen, oxygen, and nitrogen the mole fractions of the components of the combustion products are-

Mole fraction of N_2 $$X_{2,N2} = \frac{X_{1,N2}}{1 + \lambda + w} \tag{3.22}$$

Mole fraction of O_2 $$X_{2,O2} = \frac{X_{1,O2} - 2\lambda}{1 + \lambda + w} \tag{3.23}$$

Mole fraction of CO_2 $$X_{2,CO2} = \frac{X_{1,CO2} + \lambda}{1 + \lambda + w} \tag{3.24}$$

Mole fraction of H_2O $$X_{2,H2O} = \frac{X_{1,H2O} + 2\lambda + w}{1 + \lambda + w} \tag{3.25}$$

where w is the steam injection ratio defined as the ratio of mass of steam injected to the mass of air formed.

$$w = m_s/m_a \tag{3.26}$$

$$w = \left(\frac{m_s}{m_g}\right)(\frac{m_g}{m_a})$$

$$w = w^{'}(1+\lambda)$$

$$w' = w/(1+\lambda) \tag{3.27}$$

where $w^{'}$ is the ratio of mass of steam injected to the mass of gases formed.

The molar analysis of the combustion products is fixed once the fuel-air ratio λ has been determined. The fuel-air ratio can be obtained from an energy rate balance as follows.

$$0 = \dot{Q}_{cv} - \cancel{\dot{W}}_C + n_f \overline{h_f} + n_a \overline{h}_a - n_p \overline{h_p}$$

$$0 = \dot{Q}_{cv} + n_f \overline{h_f} + n_a \overline{h}_a - n_p \overline{h_p} \tag{3.28}$$

By assumption 5 of the model, the heat transfer rate is given by

$$\dot{Q}_{CV} = -l_f n_f \eta_{CC} \overline{LHV}$$

where

l_f = percentage of heat loss from combustion chamber (= 0.02)

$\overline{LHV}$ = lower heating value of fuel (802361 kJ/kmol) and

η_{CC} =combustion chamber efficiency (100%)

$$= n_a\left(-l_f \lambda \eta_{CC} \overline{LHV}\right) \tag{3.29}$$

Using ideal gas mixture principles, the enthalpies of air and combustion products are

$$h_a = \left[X_{1,N2}N_2 + X_{1,O2}O_2 + X_{1,CO2}CO_2 + X_{1,H2O}H_2O\right](T3) + wh_{s,T\,\sup}$$

(3.30)

$$(1+\lambda+w)h_p = \begin{bmatrix} X_{1,N2}h_{N_2} + (X_{1,O2}-2\lambda)h_{O_2} + (X_{1,CO2}+\lambda)h_{CO_2} \\ +(X_{1,H2O}+2\lambda+w)h_{H_2O} \end{bmatrix}(T_4)$$

(3.31)

Combining the equations from (3.26) to (3.29) and solving for λ yields

$$\lambda = \frac{X_{1,N2}\Delta h_{N_2} + X_{1,O2}\Delta h_{O_2} + X_{1,CO2}\Delta h_{CO_2} + X_{1,H2O}\Delta h_{H_2O} + w(h_{s,T\,\sup} - h_{s,T4})}{h_p - 0.02\eta_{CC}\overline{LHV} + \left(2h_{O_2} + h_{CO_2} + 2h_{H_2O}\right)(T_4)}$$

(3.32)

In case of cogeneration without steam injection, the term w vanishes and the term λ converges to the expression given in ref **[1]**.

Mass flow rate of air is given by

$$\dot{m}_a = \frac{M_a\dot{W}_{net}}{(1+\lambda+w)(\bar{h}_4 - \bar{h}_5) + (\bar{h}_1 - \bar{h}_2)}$$ (3.33)

and mass flow rate of fuel

$$\dot{m}_f = \lambda\frac{M_f}{M_a}\dot{m}_a$$ (3.34)

where M_f = Molecular weight of fuel (Methane) (16.043 kg/kmolK)

M_a = Molecular weight of air (28.649 kg/kmol)

Fuel (methane) is injected in combustion chamber at pressure p_f and temperature T_1.

Physical exergy of fuel (methane) at state f-

$$\dot{E}_f^{PH} = \dot{m}_f \left[(\overline{h}_f - \overline{h}_0) - T_0 \left(\overline{s}_f - \overline{s}_0 \right) \right] \tag{3.35}$$

Since $T_O = T_1$, using ideal gas equation, the above equation reduces to

$$\dot{E}_f^{PH} = \dot{m}_f R T_1 \ln \frac{p_f}{p_1} \tag{3.36}$$

where R = characteristics gas constant of fuel ($\bar{R} / M_f$)

Chemical exergy of fuel (methane) at state f-

$$\dot{E}_f^{CH} = \dot{m}_f \times \frac{e_{methane}^{CH4}}{M_f} \times 10^{-3} \tag{3.37}$$

where $e_{methane}^{CH4}$= standard chemical exergy of methane (824348 kJ/kmol)

Total exergy at state f can be similarly calculated as state 1.

Physical exergy at state 4-

$$\dot{E}_4^{PH} = \dot{m}_p \frac{(\overline{h_4} - \overline{h_0}) T_0 (\overline{s_4} - \overline{s_0})}{M_p} \tag{3.38}$$

At state 4, there are combustion products (flue gases) after combustion.

The restricted dead state corresponding to the mixture at state 4 consists of liquid water phase and a gas phase with mole fractions $x_{N2}^{'}$, $x_{O2}^{'}$, $x_{CO2}^{'}$, $x_{H2O,(g)}^{'}$. The contribution of the liquid water to the chemical exergy for state 4 is determined for 45 kJ per kmol of liquid water. The contribution of the gas phase to the chemical exergy is evaluated as

$$\sum (x'_k \times e_k^{CH}) + R \ To \sum (x'_k \times \ln x'_k) \tag{3.39}$$

On the basis of 1 kmol of mixture at state 4, we have $(x_{1,N2} + x_{1,O2} + x_{1,CO2} + x_{1,H2O})$ kmol as a gas phase and $[1 - (x_{1,N2} + x_{1,O2} + x_{1,CO2} + x_{1,H2O})]$ kmol as liquid water where $x_{1,N2}, x_{1,O2}, x_{1,CO2}$ and $x_{1,H2O}$ is the initial mole fraction of air before combustion, thus

Chemical exergy at state 4-

$$\dot{E}_4^{CH} = (\dot{m}_p / M_p) \times [(x_{1,N2} + x_{1,O2} + x_{1,CO2} + x_{1,H2O}) \times \sum(x'_k \times e_k^{CH}) + \bar{R}\ To\ \sum(x'_k\ ln x'_k) \times (1 - (x_{1,N2} + x_{1,O2} + x_{1,CO2} + x_{1,H2O})) \times e_w] \times 10^{-3}$$

(3.40)

where $\bar{R}$ = universal gas constant (8.314 kJ/kmolK)
Total exergy at state 4 can be similarly calculated as state 1.

Exergy destruction in combustion chamber-

$$\dot{E}_{D,CC} = \sum_i \cancel{\left(1-\frac{T_0}{T_1}\right)}\dot{Q}_1 - \cancel{\dot{W}_C} + \dot{E}_3 + \dot{E}_f - \dot{E}_4$$

$$\dot{E}_{D,CC} = \dot{E}_3 + \dot{E}_f - \dot{E}_4 \qquad (3.41)$$

3.3.4 Turbine

The purpose of using a turbine is to generate power from expansion of a gas.

After combustion the molar composition of gas changes to

$$\left[X_{2,N2}N_2 + X_{2,O2}O_2 + X_{2,CO2}CO_2 + X_{2,H2O}H_2O\right]$$

(3.42)

where

$X_{2,N2}$ = mole fraction of N_2 after combustion (= 0.7507)

$X_{2,O2}$ = mole fraction of O_2 after combustion (= 0.1373)

$X_{2,CO2}$ = mole fraction of CO_2 after combustion (= 0.0314)

$X_{2,H2O}$ = mole fraction of H_2O after combustion (= 0.0806)

The value of specific enthalpy $\overline{h}_5$ can be evaluated using the turbine isentropic efficiency. Solving the expression for expansion for turbine isentropic efficiency

$$\eta_{ST} = \frac{\overline{h}_4 - \overline{h}_5}{\overline{h}_4 - \overline{h}_{5S}} \tag{3.43}$$

we get
$$\overline{h}_5 = \overline{h}_4 - \eta_{ST}\left(\overline{h}_4 - \overline{h}_{5S}\right) \tag{3.44}$$

where η_{ST} is isentropic efficiency of turbine, $\overline{h}_4$ and $\overline{h}_5$ are enthalpies of flue gas at state 4 and 5. $\overline{h}_{5s}$ is the isentropic enthalpy at turbine exit.

The change in entropy of combustion gas after isentropic expansion in turbine

$$\begin{aligned}\overline{s^0{}_{5S}} - \overline{s}_4 = X_{2,N2}&\left[\overline{s}^0\left(T_{5s}\right) - \overline{s}^0\left(T_4\right) - \overline{R}\ln\frac{p_5}{p_4}\right]_{N2}\\ + X_{2,O2}&\left[\overline{s}^0\left(T_{5s}\right) - \overline{s}^0\left(T_4\right) - \overline{R}\ln\frac{p_5}{p_4}\right]_{O2}\\ + X_{2,CO2}&\left[\overline{s}^0\left(T_{5s}\right) - \overline{s}^0\left(T_4\right) - \overline{R}\ln\frac{p_5}{p_4}\right]_{CO2}\end{aligned}$$

$$+X_{2,H2O}\left[\bar{s}^{0}\left(T_{5s}\right)-\bar{s}^{0}\left(T_{4}\right)-\bar{R}\ln\frac{p_{5}}{p_{4}}\right]_{H2O}$$

$$=0 \qquad (3.45)$$

Physical exergy at state 5-

$$\dot{E}_{5}^{PH}=\dot{m}_{p}\frac{\bar{h}_{5}-\bar{h}_{0}-T_{0}(\bar{s}_{5}-\bar{s}_{0})}{M_{P}} \qquad (3.46)$$

Chemical exergy at state 5 will be same as that of state 4.

Total exergy at state 4 can be calculated similarly as state 1.

Exergy destruction in turbine-

$$\dot{E}_{D,T}=\sum_{i}\cancel{\left(1-\frac{T_{0}}{T_{1}}\right)\dot{Q}_{1}}+\dot{W}_{T}+\dot{E}_{4}-\dot{E}_{5}$$

$$\dot{E}_{D,T}=\dot{W}_{T}+\dot{E}_{4}-\dot{E}_{5} \qquad (3.47)$$

3.3.5 Heat Recovery Steam Generator

The waste heat recovery process in heat recovery steam generator is illustrated on temperature profile diagram in Figure 3.2. Water enters the boiler in the form of compressed liquid at condensate temperature T_{con}. As the water receives heat from the hot exhaust gases, it becomes saturated, starts boiling, and is superheated. On the hot side, the exhaust gases leaving the regenerator enter the steam generator and get cooled finally to the stack temperature.

For maximum heat recovery, the stack temperature should approach the acid dew point of exhaust gases, while keeping the pressure drops as well as the size of boiler within desirable limits. The factors which affect the cost and effectiveness of any HRSG are pinch point, approach point, allowable back pressure, stack temperature, steam temperature and pressure. The minimum temperature difference for heat transfer, which is known as pinch point plays an important role in identifying the optimum heat recovery and size of heat exchangers. Approach point is the difference between the saturation temperature and the temperature of water leaving the economizer; lowering approach point will increase probability of steaming in the economizer which may cause hammering and blanketing.

The gas side pinch point temperature (T_P) and economizer exit temperature are calculated, by assuming the drum saturation pressure (P_{Drum}).

$$T_P = T_{Drum} + PP \tag{3.48}$$

$$T_{ECO} = T_{Drum} - AP \tag{3.49}$$

Where PP and AP are the pinch point difference and approach point respectively.

The superheated steam generated $\dot{m}_{s,sup}$ for each kg/s of exhaust gases can be determined by applying the mass and energy conservation principles across superheater and economizer as-

$$\dot{m}_{s,sup} \, (\bar{h}_{s,sup} - \bar{h}_{con}) = \dot{m}_p (\bar{h}_6 - \bar{h}_7) \tag{3.50}$$

where $\bar{h}_{s,sup}$ is the enthalpy of superheated steam generated in HRSG, $\bar{h}_{con}$ is enthalpy of condensate water at inlet to HRSG, $\dot{m}_p$ is mass flow rate of gases, $\bar{h}_6$ and $\bar{h}_7$ is enthalpy of flue gases at inlet and exit of HRSG respectively.

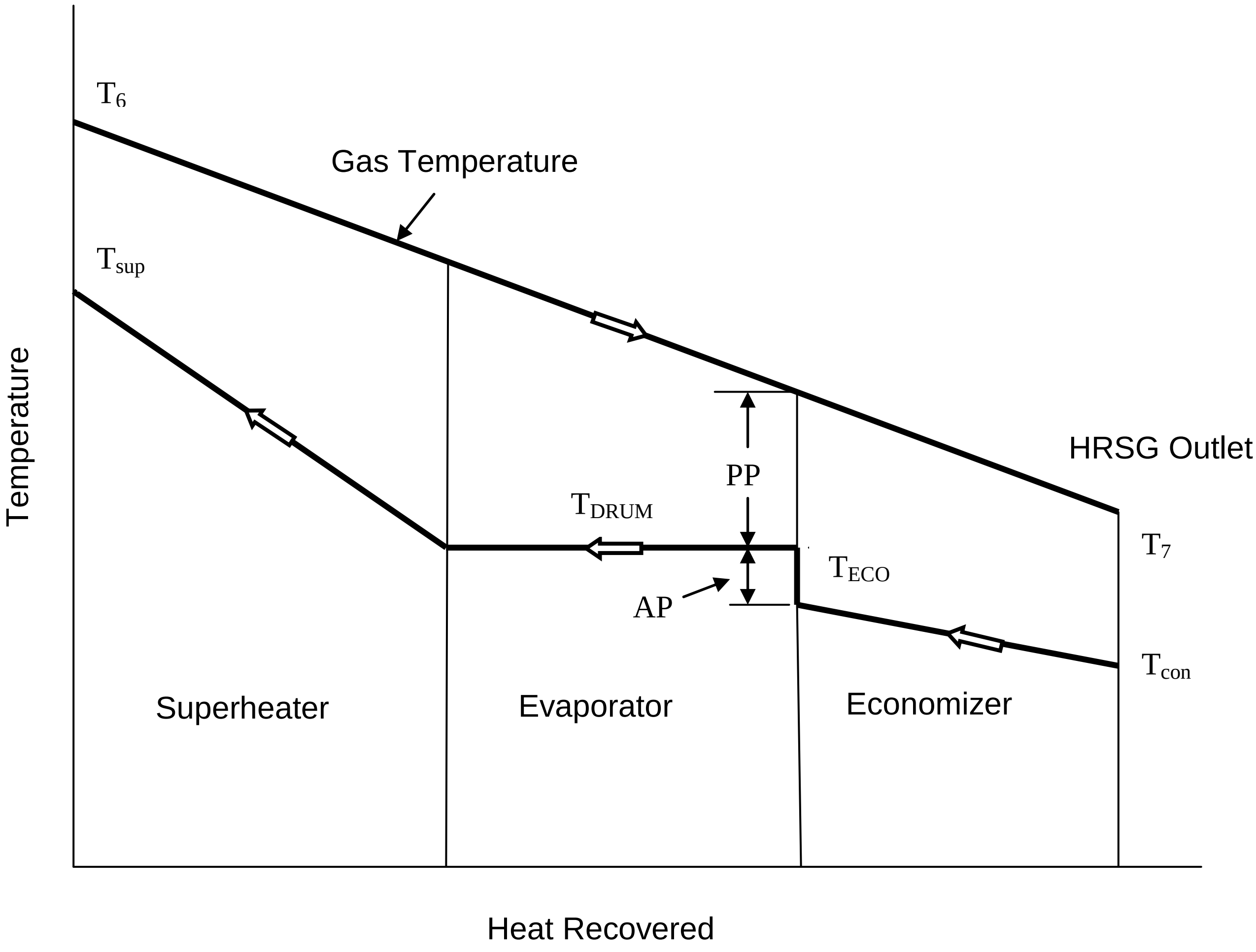

Fig. 3.2- Temperature / Heat energy diagram for HRSG

Physical exergy at state 8-

$$\dot{E}_8^{PH} = \dot{m}_8\left[h_8 - h_0 - T_0\left(s_8 - s_0\right)\right] \quad (3.51)$$

Physical exergy at state 9-

$$\dot{E}_9^{PH} = \dot{m}_9\left[h_9 - h_0 - T_0\left(s_9 - s_0\right)\right] \quad (3.52)$$

Chemical exergy at state 9-

$$\dot{E}_9{}^{CH} = \dot{m}_s \times \frac{e_w}{M_w} \times 10^{-3} \quad (3.53)$$

where e_w =standard chemical exergy of water (45 kJ/kmol)

Chemical Exergy at state 8 will be same as that of state 9.

Total exergy at state 8 and 9 can be calculated similarly as state 1.

Exergy destruction in HRSG-

$$\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_8 - \dot{E}_9 \tag{3.54}$$

The components of vapour absorption refrigeration system is discussed below-

3.3.6 Generator

The saturated steam generated in HRSG enters the generator of vapour absorption refrigeration system where it gives heat to the solution in generator. The water (refrigerant) is boiled off from the steam coils in the generator.

The heat transfer equations are as follows-

$$\dot{Q}_{GEN} = \dot{n}_{s,sat}(\bar{h}_{9'} - \bar{h}_{10}) \tag{3.55}$$

$$\dot{Q}_{GEN} = \left(\bar{h}_{17} - \bar{h}_{11}\right) + f_c(\bar{h}_{17} - \bar{h}_{16}) \tag{3.56}$$

$$f_c = [\{1 - \left(1 - \frac{X_{ws}}{100}\right)\}/\{\left(1 - \frac{X_{ss}}{100}\right) - \left(1 - \frac{X_{ws}}{100}\right)\}] \tag{3.57}$$

$$\dot{Q}_{GEN} = \dot{m}_r \ q_{GEN} \tag{3.58}$$

Here, $\dot{Q}_{GEN}$ is the heat given by the saturated steam to the generator, f_c is circulation ratio, $\dot{m}_r$ is the mass flow rate of refrigerant, q_{GEN} is specific heat of solution in the refrigerant circuit, X_{ss} is the concentration of LiBr in strong water concentration LiBr solution and X_{ws} is the concentration of LiBr in weak water concentration LiBr solution.

The exergy destruction in the generator is given by

$$\dot{E}_{D,GEN} = \dot{E}_{16} + \dot{E}_{9'} - \dot{E}_{11} - \dot{E}_{17} - \dot{E}_{10} \tag{3.59}$$

where $\dot{E}_{9'}$ and $\dot{E}_{10}$ is the exergy of saturated steam generated in the HRSG at the generator inlet and outlet.

$$\dot{E}_{16}^{PH} = \dot{m}_1[(\bar{h}_{16} - \bar{h}_o) - T_o(\bar{s}_{16} - \bar{s}_0)] \tag{3.60}$$

$$\dot{E}_{16}^{CH} = \dot{m}_1[1 - \left(\frac{X_{14}}{100}\right)]\left(\frac{1}{M_W}\right) \times e_w \tag{3.61}$$

$$\dot{E}_{16} = \dot{E}_{16}^{PH} + \dot{E}_{16}^{CH} \tag{3.62}$$

where $\dot{m}_1$ is the mass flow rate of strong solution of water from absorber, M_W is the molecular weight of water and e_w is the standard chemical exergy of water. The value of $\bar{s}_{16}$ is determined by the formula given in appendix III. In the similar way, $\dot{E}_{11}$ will be calculated.

$$\dot{E}_{17}^{PH} = \dot{m}_r[(\bar{h}_{17} - \bar{h}_o) - T_o(\bar{s}_{17} - \bar{s}_0] \tag{3.63}$$

$$\dot{E}_{17}^{CH} = \dot{m}_r\left(\frac{1}{M_W}\right)e_w \tag{3.64}$$

$$\dot{E}_{17} = \dot{E}_{17}^{PH} + \dot{E}_{17}^{CH} \tag{3.65}$$

3.3.7 Condenser

The refrigerant (high pressure water vapour) is condensed over cooling coils in the condenser and exchanges heat with water which passes through the condenser.

The heat transfer equation for condenser is given below-

$$\dot{Q}_{COND} = \dot{m}_r[(\bar{h}_{17} - \bar{h}_{18}) \tag{3.66}$$

Mass flow rate of cooling water entering the condenser may be found out by the following relation-

$$\dot{m}_{w,COND} = \dot{Q}_{COND}/[C_{pw}(T_{22} - T_{21}) \tag{3.67}$$

where T_{21} and T_{22} is inlet and exit temperature of water passing through the condenser and C_{pw} is the specific heat of water.

The exergy destruction in the condenser is given by

$$\dot{E}_{D,COND} = (\dot{E}_{17} - \dot{E}_{18}) + (\dot{E}_{21} - \dot{E}_{22}) \tag{3.68}$$

$$\dot{E}_{18}^{PH} = \dot{m}_r[(\bar{h}_{18} - \bar{h}_o) - T_o(\bar{s}_{18} - \bar{s}_0] \tag{3.69}$$

$$\dot{E}_{18}^{CH} = \dot{m}_r\left(\frac{1}{M_W}\right)e_w \tag{3.70}$$

$$\dot{E}_{18} = \dot{E}_{18}^{PH} + \dot{E}_{18}^{CH} \tag{3.71}$$

3.3.8 Evaporator

The condensed refrigerant (condensate) from condenser is supplied through an expansion valve into evaporator where it gets heated from the circulating water and is converted into low pressure water vapour. Cooled water in the evaporator is sent to the air heat exchanger in which air at ambient condition is passed and gets cooled by the chilled water. The cooled air in then sent to compressor of gas turbine unit.

The heat transfer equation for evaporator is given below-

$$\dot{Q}_{EVAP} = \dot{m}_r[(\bar{h}_{20} - \bar{h}_{19}) \tag{3.72}$$

The exergy destruction in the evaporator is given by

$$\dot{E}_{D,EVAP} = (\dot{E}_{19} - \dot{E}_{20}) + (\dot{E}_{21} - \dot{E}_{22}) \tag{3.73}$$

$$\dot{E}_{19}^{PH} = \dot{m}_r[(\bar{h}_{19} - \bar{h}_o) - T_o(\bar{s}_{19} - \bar{s}_0] \tag{3.74}$$

$$\dot{E}_{19}^{CH} = \dot{m}_r \left(\frac{1}{M_W}\right) e_w \tag{3.75}$$

$$\dot{E}_{19} = \dot{E}_{19}^{PH} + \dot{E}_{19}^{CH} \tag{3.76}$$

In the similar fashion, $\dot{E}_{20}$ will be calculated.

3.3.9 Absorber of VARS

The low pressure water vapour generated in evaporator is further fed into the absorber. The weak concentration solution of water in LiBr remaining in the generator is also fed back to the absorber through the solution heat exchanger and throttling valve. The weak solution stream is mixed with the refrigerant vapour and form a strong solution which is pumped by the solution pump to generator after getting heated in the heat exchanger.

The heat transfer equation for absorber is given below-

$$\dot{Q}_{ABS} = \dot{m}_r\left(\bar{h}_{20} - \bar{h}_{13}\right) + f_c(\bar{h}_{13} - \bar{h}_{14}) \tag{3.77}$$

Mass flow rate of cooling water entering the absorber may be found out by the following relation-

$$\dot{m}_{w,ABS} = \dot{Q}_{ABS}/[C_{pw}(T_{24} - T_{23}) \tag{3.78}$$

The exergy destruction in the absorber is given by

$$\dot{E}_{D,ABS} = (\dot{E}_{20} + \dot{E}_{13} - \dot{E}_{14}) + (\dot{E}_{23} - \dot{E}_{24}) \tag{3.79}$$

Since $\dot{E}_{12}$ = $\dot{E}_{13}$, the above equation may be written as

$$\dot{E}_{D,ABS} = (\dot{E}_{20} + \dot{E}_{12} - \dot{E}_{14}) + (\dot{E}_{23} - \dot{E}_{24}) \tag{3.80}$$

$$\dot{E}_{12}^{PH} = \dot{m}_2[\left(\bar{h}_{12} - \bar{h}_o\right) - T_o(\bar{s}_{12} - \bar{s}_0] \tag{3.81}$$

$$\dot{E}_{12}^{CH} = \dot{m}_2 \left[1 - \left(\frac{X_{11}}{100}\right)\right] \left(\frac{1}{M_W}\right) e_w \tag{3.82}$$

$$\dot{E}_{12} = \dot{E}_{12}^{PH} + \dot{E}_{12}^{CH} \tag{3.83}$$

The value of $\bar{s}_{12}$ is determined by the formula given in appendix III.

In the similar way,

$$\dot{E}_{14}^{PH} = \dot{m}_1 [(\bar{h}_{14} - \bar{h}_o) - T_o (\bar{s}_{14} - \bar{s}_0] \tag{3.84}$$

$$\dot{E}_{14}^{CH} = \dot{m}_1 \left[1 - \left(\frac{X_{14}}{100}\right)\right] \left(\frac{1}{M_W}\right) e_w \tag{3.85}$$

$$\dot{E}_{14} = \dot{E}_{14}^{PH} + \dot{E}_{14}^{CH} \tag{3.86}$$

The value of $\bar{s}_{14}$ is determined by the formula given in appendix III.

3.3.10 Solution Heat Exchanger (SHX)

In solution heat exchanger, the weak solution in hot state from generator enters and heat up the strong solution which enters in cold state from absorber.

The heat exchange between two solutions is given by the following equation-

$$\dot{Q}_{SHX} = \dot{m}_r f_c (\bar{h}_{16} - \bar{h}_{15}) \tag{3.87}$$

$$\text{also } \dot{Q}_{SHX} = \dot{m}_r (f_c - 1)(\bar{h}_{16} - \bar{h}_{15}) \tag{3.88}$$

$$\dot{m}_1 = \dot{m}_r f_c \tag{3.89}$$

$$\dot{m}_2 = \dot{m}_r (f_c - 1) \tag{3.90}$$

where $\dot{m}_1$ is the mass flow rate of strong solution of water from absorber and $\dot{m}_2$ is mass flow rate of weak solution of water from generator.

The value of enthalpy at state 16 may be found out by the following relation-

$$h_{16} = h_{15} + [(f_c - 1)/f_c](\bar{h}_{11} - \bar{h}_{12}) \tag{3.91}$$

The exergy destruction in the solution heat exchanger is given by

$$\dot{E}_{D,SHX} \ = \ \dot{E}_{15} - \dot{E}_{16} - \dot{E}_{11} - \dot{E}_{12} \qquad (3.92)$$

$$\dot{E}_{15}^{PH} \ = \dot{m}_1[(\bar{h}_{15} - \bar{h}_o) - T_o(\bar{s}_{15} - \bar{s}_0] \qquad (3.93)$$

$$\dot{E}_{14}^{CH} \ = \dot{m}_1 \left[1 - \left(\frac{X_{14}}{100}\right)\right] \left(\frac{1}{M_W}\right) e_w \qquad (3.94)$$

$$\dot{E}_{15} \ = \dot{E}_{15}^{PH} + \dot{E}_{14}^{CH} \qquad (3.95)$$

The value of $\bar{s}_{15}$ is determined by the formula given in appendix III.

$$\dot{E}_{11}^{PH} \ = \dot{m}_2[(\bar{h}_{11} - \bar{h}_o) - T_o(\bar{s}_{11} - \bar{s}_0] \qquad (3.96)$$

$$\dot{E}_{11}^{CH} \ = \dot{m}_2 \left[1 - \left(\frac{X_{11}}{100}\right)\right] \left(\frac{1}{M_W}\right) e_w \qquad (3.97)$$

$$\dot{E}_{11} \ = \dot{E}_{11}^{PH} + \dot{E}_{11}^{CH} \qquad (3.98)$$

The value of $\bar{s}_{11}$ is determined by the formula given in appendix III.

3.3.11 Solution Pump

The function of the solution pump is to raise the solution to a higher pressure in a generator from a low pressure maintained in the absorber.

The work consumed by the pump is given by following equation

$$\dot{W}_{pump} \ = [v_{14}(P_{GEN} - P_{ABS})\dot{m}_1/\eta_{pump} \ ] \qquad (3.99)$$

$$= \dot{} m_1[(\bar{h}_{15} - \bar{h}_{14}) \qquad (3.100)$$

where η_{pump} is the efficiency of pump, P_{GEN} & P_{ABS} is the pressure of generator and absorber& v is the specific volume of strong solution entering the pump.

3.3 Performance Parameters

The relevant parameters required for the thermodynamic analysis of gas turbine cogeneration system integrated with IAC and STIG are summarized below:

3.3.1 First–Law Efficiency (η_I) - The ratio of all the useful energy extracted from the system (electricity and process heat) to the energy of fuel input is known as first-law efficiency. First-law efficiency is also known as fuel utilization efficiency or energetic efficiency. By definition,

$$\eta_I = (\dot{W}_{el} + \dot{Q}_{EVAP} + \dot{Q}_{Pro})/\dot{Q}_f \qquad (3.101)$$

where $\dot{W}_{el}$ is electrical power output, $\dot{Q}_{EVAP}$ is refrigerating capacity, $\dot{Q}_{Pro}$ is process heat rate and $\dot{Q}_f$ is the heat rate supplied by the fuel.

3.3.2 Power to heat ratio ($\boldsymbol{R_{PH}}$) - The cost effectiveness of any cogeneration system is directly related to the amount of power it can produce for a given amount of process heat added. Hence, another parameter used to assess the thermodynamic performance of such a cogeneration system is R_{PH}, which is defined as

$$R_{PH} = \dot{W}_{el}/(\dot{Q}_{EVAP} + \dot{Q}_{Pro}) \qquad (3.102)$$

In both the first-law efficiency and power-to-heat ratio, power and heat are treated as equal. This reflects the first law of thermodynamics, which is concerned with quantity, not energy quality. Thus η_I and R_{PH} are also known as first-law efficiency.

3.3.3 Second–Law Efficiency (η_{II}) – Since exergy is more valuable than energy according to the second law of thermodynamics, it is useful to consider both output and input in terms of exergy. The amount of exergy supplied in the product to the amount of exergy associated with the fuel is a more accurate measure of thermodynamic performance of a system, which is called second-law efficiency. It is also called exergetic efficiency (effectiveness or rational efficiency). By definition,

$$\eta_{II} = (\dot{W}_{el} + \dot{E}_{EVAP} + \dot{E}_{Pro})/\dot{E}_f \qquad (3.103)$$

where $\dot{E}_{EVAP}$ is the exergy content of cold in evaporator, $\dot{E}_{Pro}$ is the exergy content of process heat and $\dot{E}_f$ is the exergy content of fuel input.

3.3.4 Exergy Destruction Ratio (y_D) - It can be defined as the ratio of rate of exergy destruction in a system component to the exergy rate of the fuel provided to the overall system.

$$y_D = \dot{E}_D / \dot{E}_{f,tot} \qquad (3.104)$$

Alternatively, the component exergy destruction rate can be compared to the total exergy destruction rate within the system.

$$\dot{y}_D = \dot{E}_D / \dot{E}_{D,tot} \qquad (3.105)$$

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

3.3.5 Exergy Loss Ratio (y_L) - It can be defined as the ratio of rate of exergy loss in a system component to the exergy rate of the fuel provided to the overall system.

$$y_L = E_L / E_{f,tot} \qquad (3.106)$$

3.3.6 Pinch Point (PP) - Pinch is used in sizing the heat transfer surface area of HRSG. As seen in Fig. 3.2, the pinch point is the difference between the saturation temperature and the temperature of flue gases at particular point along the gas path. It is desirable to make pinch point as small as possible. The minimum temperature difference for heat transfer, which is known as pinch point plays an important role in identifying the optimum heat recovery and size of heat exchangers. HRSG'S are popular today because of the efficiency gains that are accomplished with the reuse of waste heat from turbines.

3.3.7 CHP Electric Effectiveness or Energy Utilization Factor (EUF) - Another parameter used to evaluate the cogeneration system performance is energy utilization factor. For cogeneration electrical power and heat, the energy utilization factor is given by

$$\varepsilon_{EE} = W_{el} / (Q_f - \left(\frac{Q_p}{\eta_{HRSG}}\right)$$

Where

W_{el} = Electric power output, kW

Q_f = Heat supplied by fuel based on LHV, kW

Q_p = Process heat supplied to process steam, kW

3.3.8 Fuel Energy Saving Ratio (FESR) - It is known that the production of work is more difficult than the heat, which is a cheap product of a station. Hence the utilization factor does not satisfy completely as characteristics factor since it gives the same value at work and heat. Another proposed criterion for the effectiveness of cogeneration systems is the fuel energy savings ratio (FESR), which takes into account the consumption of fuel. FESR can be represented as

$$\emptyset_{saving} = (FUEL_{reference} - Q_f)/\, FUEL_{reference}$$

Here

$$FUEL_{reference} = (\frac{W_{el}}{\eta_g}) + (Q_p/\eta_{HRSG})$$

CHAPTER 4
SIMULATION STUDIES

This chapter describes the formulation of gas turbine cogeneration system integrated with IAC and STIG. Based upon this formulation, solution methodology, model validation, comparison of results and discussions are given below.

4.1 Solution Methodology

The structure of computer program consists of different equations written in EES (Engineering Equation Solver) software to calculate wide range of output parameters required. These parameters have been used to perform energy and exergy analysis of different components of the system. Different properties (fuel, air, combustion products etc.) are inbuilt in EES software which has been used directly in calculation. One of the most interesting features of EES is that it can solve the problem iteratively in less time and the solution of required output parameters appears in the solution window. The main condition of solving any problem is that the number of equations should be equal to number of unknowns.

4.2 Model Validation

Initially, the computer program developed for cogeneration system has been validated with the system given in ref **[1]**. The various components of the gas turbine cogeneration system with regeneration is shown in Figure 4.1. In this system, regenerator has been used between compressor and combustion chamber. The program made in EES software can be used for systems with regeneration or without regeneration.

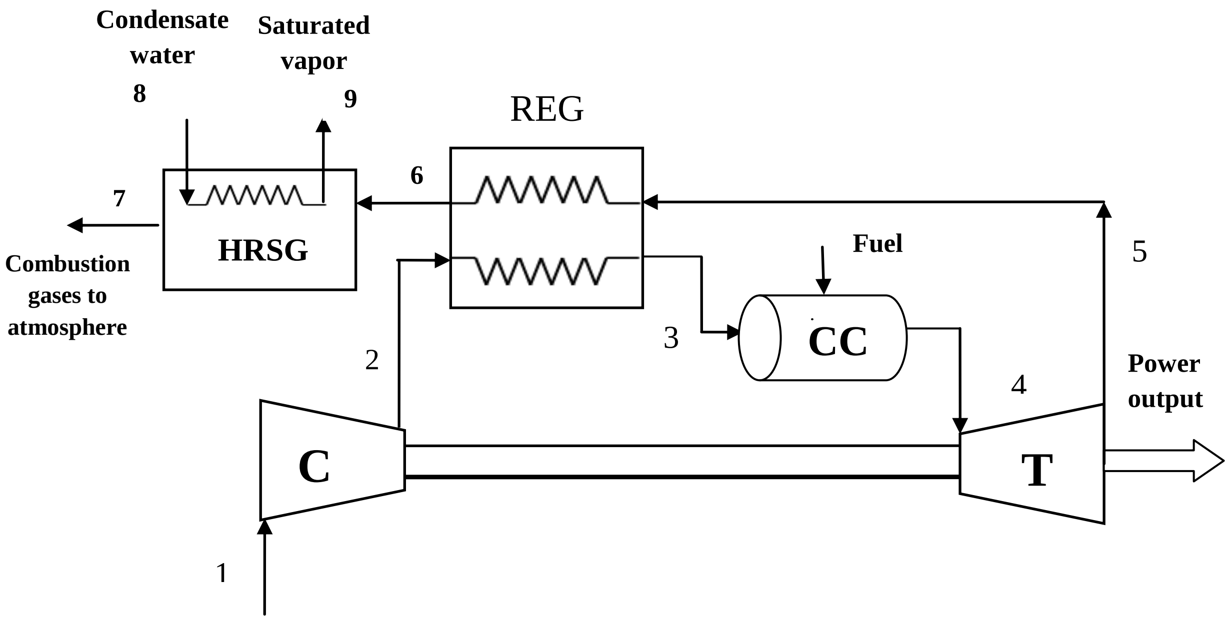

Fig. 4.1 Gas turbine cogeneration system with regeneration, ref **[1]**

The various input parameters taken from ref **[1]** supplied to the program are given in Table 4.1.

Table 4.1 Input data for analysis of cogeneration plant

Air inlet pressure to compressor (P_1), in bar	1.013
Air inlet temperature to compressor, (T_1) in K	298.15
Pressure ratio of compressor (r_p)	10:1
Isentropic efficiency of compressor (η_{SC}), in %	0.86
Turbine inlet temperature (TIT) or maximum cycle temperature (T_4), in K	1520
Isentropic efficiency of Turbine (η_{ST}), in %	0.86
Exhaust pressure of combustion prod after HRSG (P_7), in bar	1.013
Mass flow rate of steam generated in HRSG (m_s) in kg/s	14
Pressure of steam generation (P_9) in bar	20
Pressure of condensate water at inlet of HRSG (P_8), in bar	20
Temperature of condensate water at inlet to HRSG (T_8), in K	298.15
Lower heating value of fuel (LHV), in kJ/kmol	802361
Net power output of the plant (W_{net}), in MW	30
Injection pressure of fuel (methane) (P_f), in bar	12
Injection temperature of fuel (methane) (T_f), in K	298.15
Pressure drop in regenerator on the gas side , in %	3
Pressure drop in regenerator on the air side, in %	5
Pressure drop in combustion chamber , in %	5
Pressure drop in HRSG on the gas side, in %	5

Table 4.2 to 4.3 represent the comparison of simulated values with ref. **[1]** for operating parameters, calculated parameters and exergy destruction data. The bracket terms represent value of ref **[1].**

Table 4.2 Comparison of operating parameters with ref **[1]**

State	**Substance**	**Mass flow rate (kg/s)**	**Temperature (K)**	**Pressure (bar)**
1	Air	90.74 (91.27)	298.15	1.013
2	Air	90.74 (91.27)	611.2 (603.738)	10.13 (10.130)
3	Air	90.74 (91.27)	850	9.624 (9.623)
4	Combustion Products	92.38 (92.91)	1520	9.142 (9.142)
5	Combustion products	92.38 (92.91)	1010 (1006.162)	1.099 (1.099)
6	Combustion products	92.38 (92.91)	793.2 (779.884)	1.066 (1.066)
7	Combustion products	92.38 (92.91)	427.8 (426.897)	1.013
8	Water	14	298.15	20
9	Steam	14	485.4 (485.570)	20
f	Methane	1.642 (1.641)	298.15	12

Table 4.3 Comparison of calculated parameters with ref. **[1]**

State	Substance	Enthalpy (kJ/kmol)	Entropy (kJ/kmolK)	Physical exergy (MW)	Chemical exergy (MW)	Total exergy (MW)
1	Air	-4713 (-4713.30)	199.1	0	0	0
2	Air	4632 (4596.9)	201.3	27.504 (27.5382)	0	27.504 (27.5382)
3	Air	12190 (12524)	212.2 (212.90)	41.204 (41.9384)	0	41.204 (41.9384)
4	Combustion Products	8963 (9304.41)	235 (235.70)	100.206 (101.0873)	0.6046 (0.3665)	100.811 (101.453)
5	Combustion products	-9264 (-8839)	238.1	37.650 (38.4158)	0.6046 (0.3665)	38.255 (38.7823)
6	Combustion products	-16586 (-16522.68)	230.2	21.408 (21.3851)	0.6046 (0.3665)	22.012 (21.7516)
7	Combustion products	-28123 (-27974.5)	211.2	2.172 (2.4061)	0.6046 (0.3665)	2.776 (2.7726)
8	Water	1920 (1884.6)	6.601	0.02667 (0.0266)	0.035 (0.0350)	0.06167 (0.0616)
9	Steam	50420 (50.347.8)	114.2	12.807 (12.7752)	0.035 (0.0350)	12.842 (12.8102)
f	Methane	-74872	-	0.6267 (0.6267)	84.35 (84.3668)	84.979 (84.9939)

Table 4.4 Comparison of exergy destruction data with ref **[1]**

Component	Exergy Destruction Rate (MW)	Exergy Destruction (Percentage a)	Exergy Destruction (Percentage b)
Combustion chamber	25.37 (25.48)	64.36 (64.56)	29.85 (29.98)
HRSG	6.46 (6.23)	16.37 (15.78)	7.59 (7.33)
Gas turbine	2.96 (3.01)	7.50 (7.63)	3.48 (3.54)
Air preheater	2.54 (2.63)	6.44 (6.66)	2.99 (3.09)
Air compressor	2.09 (2.12)	5.30 (5.37)	2.46 (2.49)
Overall plant	39.42 (39.47)	100.00	46.39 (46.43)

Percentage a ($\%^a$) is the exergy destruction rate within a component as a percentage of the total exergy destruction rate within the cogeneration system.

Percentage b ($\%^b$) is the exergy destruction rate within a component as a percentage of the exergy rate entering the cogeneration system with the fuel.

From the above results, it can be concluded that the output data of present model matches reasonably well with the corresponding values of ref **[1]** given in the bracket.

4.2.1 Case Study

In the study, a simple cycle 7B Frame GENSET of is considered as the base unit. The rated output of this unit under ISO condition (101.3 kPa, 288 K, 60% RH) is 60.3 MW if natural gas is used and 59.0 MW if distillate oil is used. In order to recover the energy from the exhaust gases, a (HRSG) is installed to recapture most of this energy and generate steam with pressure suitable for a VAR system or/and for a steam injection system. at the downstream exit of the turbine. The maximum energy that can be recovered by the HRSG is limited by the effectiveness of the HRSG and the outlet temperature of the flue gas. The effectiveness of the HRSG is set to be 0.8, and the exhaust from the stack is taken as 130˚C to prevent the possibility of vapour condensation

The system component data and input conditions of this unit have been summarized in Table 4.5 ref **[11]**.

In this simulation, the efficiencies of compressor and turbine were taken as 0.86 and 0.87, respectively. The cogeneration system considered in the case study is given in Figure 4.2.

Following three different cases of the system has been analyzed-

1. Gas turbine cogeneration system (without STIG)
2. Gas turbine cogeneration system (with STIG)
3. Gas turbine cogeneration system (with IAC)

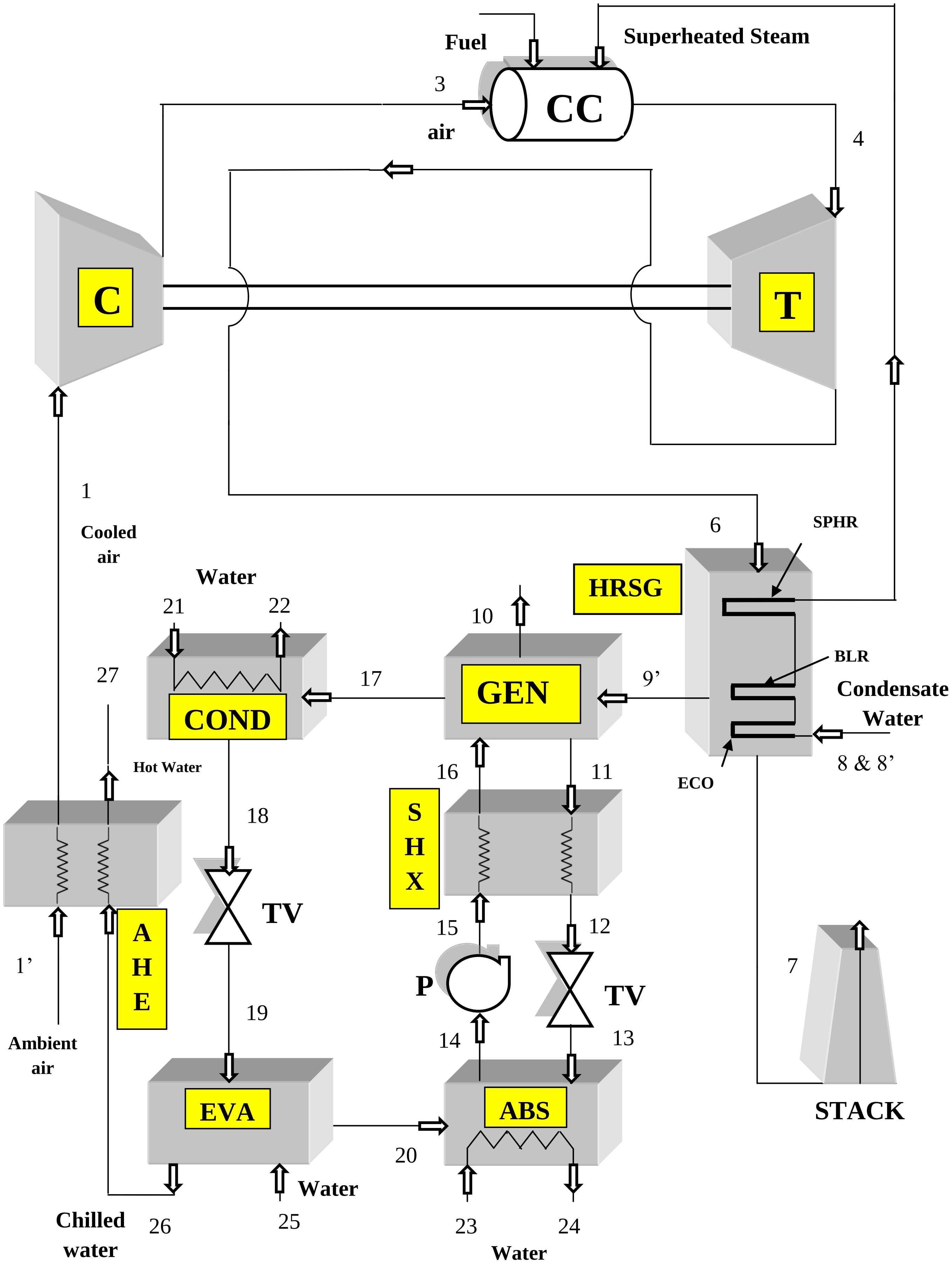

Fig. 4.2 Cogeneration system integrated with IAC and STIG

4.2.1.1 Gas Turbine Cogeneration System (without STIG)

The gas turbine cogeneration system consists of compressor, combustion chamber, gas turbine and HRSG.

The various input parameters supplied to the program are given in Table 4.5. Some parameters like HRSG exhaust pressure and temperature, temperature of condensate water at inlet to HRSG not given in ref **[11]** are suitably assumed.

Table 4.5 Input data for analysis of cogeneration plant without steam injection

Air inlet pressure to compressor (P_1), in bar	1.013
Air inlet temperature to compressor, (T_1) in K	305
Pressure ratio of compressor (r_p)	9
Isentropic efficiency of compressor (η_{SC}), in %	86
Turbine inlet temperature (TIT) or maximum cycle temperature (T_4), in K	1264
Isentropic efficiency of Turbine (η_{ST}), in %	87
Generator efficiency(η_G), in %	98
Exhaust pressure of combustion prod after HRSG (P_7), in bar	1.013
Pressure of steam generation (P_9) in bar	14.01
Pressure of condensate water at inlet of HRSG (P_8), in bar	14.01
Temperature of condensate water at inlet to HRSG (T_8), in K	305
Lower heating value of fuel (LHV), in kJ/kmol	802361
Mass flow rate of air (m_a), in kg/s	221.28
Injection pressure of fuel (methane) (P_f), in bar	17
Injection temperature of fuel (methane) (T_f), in K	298.15

Pressure drop in compressor , in %	1
Pressure drop in regenerator on the gas side , in %	3
Pressure drop in regenerator on the air side, in %	5
Pressure drop in combustion chamber , in %	5
Pressure drop in turbine , in %	1
Pressure drop in HRSG on the gas side, in %	5

Table 4.6 to 4.9 represent the simulated values for operating parameters, calculated parameters and exergy destruction data. The bracket terms represent values given in ref **[11]**.

Table 4.6 Comparison of operating parameters without steam injection with ref **[11]**

State	Substance	Mass flow rate (kg/s)	Temperature (K)	Pressure (bar)
1	Air	221.28	305	1.013
3	Air	221.28	604.1	9.028
4	Combustion Products	225	1264	8.57
5	Combustion products	225	827.9	1.056
6	Combustion products	225	827.9	1.067
7	Combustion products	225	393	1.013
8	Water	31.91	305	1.41
9	Steam	31.91	802.9	1.41
f	Methane	3.693	298.15	1.7

Table 4.7 Comparison of calculated parameters without steam injection with ref **[11]**

State	Substance	Enthalpy (kJ/kmol)	Entropy (kJ/kmolK)	Physical exergy (MW)	Chemical exergy (MW)	Total exergy (MW)
1	Air	-4507	199.8	0.02384	0	0.02384
3	Air	4415	201.9	64.014	0	64.014
4	Combustion Products	1656	228.7	184.092	1.241	185.333
5	Combustion products	-13353	231.6	58.018	1.241	59.259
6	Combustion products	-13353	231.6	58.018	1.241	59.259
7	Combustion products	-27120	208.6	2.973	1.241	4.214
8	Water	2438	8.351	0.06921	0.080	0.1492
9	Steam	64030	138.7	52.541	0.080	52.621
10	Methane	-74872	-	1.608	189.747	191.355

Table 4.8 Performance parameters for gas turbine cogeneration system without steam injection

Performance Parameter	**Cogeneration without Steam Injection**
First law efficiency (in %)	84.61
Second law efficiency (in %)	75.98
Power generation efficiency (%)	27.02
Power-to-heat ratio	0.4371
Fuel-air ratio	0.0298
Steam injection ratio	0
Turbine work (MW)	117.419
Compressor power (MW)	68.916
Net power output (MW	48.502

Table 4.9 Comparison of exergy destruction data without steam injection with ref **[11]**

Component	**Exergy Destruction Rate (MW)**	**Exergy Destruction (Percentage a)**	**Exergy Destruction (Percentage b)**
Combustion chamber	70.037 (69.38)	72.75	36.6
HRSG	14.618	15.19	7.639
Gas turbine	6.685 (7.16)	6.944	3.493
Air compressor	4.926 (5.10)	5.117	2.574
Overall plant	96.266	100	50.30

Percentage a (%^{a}) is the exergy destruction rate within a component as a percentage of the total exergy destruction rate within the cogeneration system.

Percentage b (%^{b}) is the exergy destruction rate within a component as a percentage of the exergy rate entering the cogeneration system with the fuel.

From the results, it can be concluded that combustion chamber is the major component of thermodynamic inefficiency. The next prominent component is the heat-recovery steam generator (HRSG). Small contribution to inefficiency is made by the gas turbine and air compressor. The exergy destructions in these components stem from one or more of three principal irreversibilities associated, respectively, with chemical reaction, heat transfer, and friction.

4.2.1.2 Gas Turbine Cogeneration System with Steam Injection

The various input parameters taken from ref **[11]** supplied to the program has been given in Table 4.5. The total steam generated in the HRSG is used for steam injection.

Table 4.10 to 4.13 represent the comparison of simulated values with ref. **[11]** for operating parameters, calculated parameters and exergy destruction data. The bracket terms represent value of ref **[11].**

Table 4.10 Comparison of operating parameters with steam injection with ref **[11]**

State	**Substance**	**Mass flow rate (kg/s)**	**Temperature (K)**	**Pressure (bar)**
1	Air	221.28	305	1.013
2	Air	221.28	604.1	9.028
3	Air	221.28	604.1	9.028
4	Combustion Products	267.6	1264	8.75
5	Combustion products	267.6	844.3	1.067
6	Combustion products	267.6	844.3	1.067
7	Combustion products	267.6	393	1.013
8	Water	41.92	305	1.41
9	Steam	41.92	824.3	1.41
f	Methane	4.442	298.15	1.7

Table 4.11- Comparison of calculated parameters with steam injection with ref **[11]**

State	**Substance**	**Enthalpy (kJ/kmol)**	**Entropy (kJ/kmolK)**	**Physical exergy (MW)**	**Chemical exergy (MW)**	**Total exergy (MW)**
1	Air	-4507	199.8	0.02384	0	0.02384
2	Air	4415	201.9	64.014	0	64.014
3	Air	4415	201.9	64.014	0	64.014
4	Combustion Products	-40607	231	234.537	9.622	244.159
5	Combustion products	-55771	233.8	77.280	9.622	86.902
6	Combustion products	-55771	233.8	77.280	9.622	86.902
7	Combustion products	-70502	209.5	3.777	9.622	13.39
8	Water	2438	8.351	0.06985	0.1048	0.1747
9	Steam	64616	139.4	53.899	0.1048	54.004
10	Methane	-74872	-	1.935	228.246	230.180

Table 4.12- Performance parameters for gas turbine cogeneration system with steam injection

Performance Parameter	**Cogeneration without Steam Injection**
First law efficiency (in %) (equivalent to power generation efficiency)	38.78 %
Second law efficiency (in %)	37.06%
Fuel-air ratio	0.03626
Steam injection ratio	0.2397
Turbine work (MW)	156.989
Compressor power (MW)	68.916
Net power output (MW)	88.073

Table 4.13- Comparison of exergy destruction data with steam injection with ref **[11]**

Component	**Exergy Destruction Rate (MW)**	**Exergy Destruction (Percentage a)**	**Exergy Destruction (Percentage b)**
Combustion chamber	104.039	76.04	45.2
HRSG	19.674	14.38	8.547
Gas turbine	8.184	5.982	3.556
Air compressor	4.926	3.6	2.14
Overall plant	136.823	100.00	59.443

Percentage a (%^{a}) is the exergy destruction rate within a component as a percentage of the total exergy destruction rate within the cogeneration system.

Percentage b (%^{b}) is the exergy destruction rate within a component as a percentage of the exergy rate entering the cogeneration system with the fuel.

From the results, it can be concluded that exergy destruction rate in the combustion chamber is highest. All the principal irreversibilities like chemical reaction, heat transfer, and friction are present in combustion chamber. Steam injection will also increase the exergy destruction due to mixing in the combustor. The total exergy loss per thermodynamic cycle increases with the addition of STIG feature because more reaction process will be implemented into the system.

4.2.1.3 Gas Turbine Cogeneration System Inlet air Cooling and process steam (Trigeneration)

The various input parameters taken from ref **[11]** supplied to the program has been given in Table 4.5. The input parameters for vapour absorption refrigeration system is given below in Table 4.14.The steam generated in the HRSG is used for inlet air cooling and remaining steam is used as process steam.

Table 4.14 Input data for analysis of cogeneration plant with inlet air cooling and process steam

Input parameters for vapour absorption refrigeration system

Generator temperature (T_g) in K	363
Absorber temperature (T_a) in K	308
Condenser temperature (T_c) in K	308
Evaporator temperature (T_e) in K	278
Temperature of water at inlet to condenser (T_{21}), in K	298
Temperature of water at outlet to condenser (T_{22}), in K	308
Temperature of water at inlet to absorber (T_{23}), in K	288
Temperature of water at outlet to absorber (T_{24}), in K	308
Temperature of water at inlet to evaporator (T_{25}), in K	288
Temperature of water at outlet to evaporator (T_{26}), in K	283
Efficiency of pump (η_{pump}) ,in %	90
Effectiveness of generator	0.8
Effectiveness of solution heat exchanger (E_{SHX})	0.9
Effectiveness of air heat exchanger (E_{AHE})	0.75

Table 4.15 to 4.18 represent the comparison of simulated values with ref. **[11]** for operating parameters, calculated parameters and exergy destruction data. The bracket terms represent value of ref **[11].** All the generated saturated steam is used for heating the generator of vapour absorption refrigeration system. Some amount of steam is used for cooling the inlet air and remaining steam is used as process heat.

Table 4.15 Comparison of operating parameters with ref **[11]** for inlet air cooling and process steam (trigeneration)

State	**Substance**	**Mass flow rate (kg/s)**	**Temperature (K)**	**Pressure (bar)**
1	Air	221.28	299.35	1.013
3	Air	221.28	593.4	9.624 (9.623)
4	Combustion Products	224.75	1264	9.142 (9.142)
5	Combustion products	224.75	829.7	1.099 (1.099)
7	Combustion products	224.75	393.0	1.013
8	Water	32	305	14
8’	Water	41.21	305	8
9	Superheated steam	32	804.7	14
9’	Saturated steam	41.12	443.4	8
f	Methane	3.749	305	12
10	Steam	41.12	403	8
11	Weak solution	69.07	353	6.243
12	Weak solution	69.07	318.65	6.243
13	Weak solution	69.07	318.65	0.93
14	Strong solution	59.08	314	0.93
15	Strong solution	59.08	314.83	6.241

16	Strong solution	59.08	392.9	6.241
17	Refrigerant	2.136	363	6.241
18	Refrigerant	2.136	308	6.241
19	Refrigerant	2.136	281	0.93
20	Refrigerant	2.136	281	0.93
21	Cooling water	9.62	298	1.013
22	Cooling water	9.62	308	1.013
23	Cooling water	14.25	288	1.013
24	Cooling water	14.25	308	1.013
25	Cooling water	8.89	305	1.013
26	Cooling water	8.89	305	1.013
27	Cooling water	8.89	296	1.013
1’	Air	221.28	299.35	1.013

Table 4.16 Comparison of calculated parameters with ref **[11]** for inlet air cooling and process steam (trigeneration)

State	Substance	Enthalpy (kJ/kmol)	Entropy kJ/kmolK	Total Exergy (MW)
1	Air	-4673	199.3	0.00567
3	Air	4086	201.4	62.738
4	Combustion Products	1297		185.530
5	Combustion products	-13715	231.6	59.390
7	Combustion products	-27491	239.6	4.277
8	Water	1911	6.604	0.125
8'	Water	1900	6.324	0.112
9	Superheated steam	63573	138.2	40.172
f	Methane	-74872	-	194.258
9'	Saturated steam	49882	36.79	92.164
10	Steam	9850	29.44	5.759
11	Weak solution	189.8	0.4645	2.474
12	Weak solution	122.7	0.2674	2.460
13	Weak solution	122.7	0.2674	2.460
14	Strong solution	94.83	0.2545	2.593
15	Strong solution	98.6	0.2663	2.594
16	Strong solution	161.2	0.7207	2.463
17	Refrigerant	47730	153.7	0.3992
18	Refrigerant	2867	9.827	0.164
19	Refrigerant	2867	12.55	0.06782
20	Refrigerant	2642	160.3	0.19384

21	Cooling water	1889	6.61	29.38
22	Cooling water	2642	9.096	29.46
23	Cooling water	1135	6.61	29.53
24	Cooling water	2462	9.096	29.84
25	Cooling water	2416	9.075	52.67
26	Cooling water	1849	9.032	49.04
27	Cooling water	2275	6.874	5.732
1’	Air	-4676	199.8	0.00584

Table 4.17 Performance parameters for gas turbine cogeneration system with inlet air cooling and process steam (trigeneration)

Performance Parameter	Inlet Air Cooling and Process Steam (Trigeneration)
First law efficiency (in %)	53.25
Second law efficiency (in %)	51.08
Power generation efficiency (%)	27.62
Fuel-air ratio	0.03692
Power-to-heat ratio	10.15
Steam injection ratio	0
Turbine work (MW)	119.45
Compressor power (MW)	67.656
Net power output (MW)	51.797
COP of VAR system	0.267
Circulation ratio (kmol/kmol of vapour)	15.2

Table 4.18 Comparison of exergy destruction data with ref **[11]** for inlet air cooling and process steam (trigeneration)

Component	**Exergy Destruction Rate (MW)**	**Exergy Destruction (Percentage a)**	**Exergy Destruction (Percentage b)**
Combustion chamber	71.466 (75.34)	70.32	36.78
HRSG	15.062 (15.95)	14.82	7.75
Gas turbine	6.687 (6.52)	6.58	3.44
Air compressor	4.923 (6.49)	4.84	2.53
VAR System	3.49 (3.27)	3.43	1.79
Overall plant	101.62	100	52.29

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Table 4.19 Exergy destruction data for VAR system

Component	**Exergy Destruction Rate (MW)**	**Exergy Destruction (Percentage a)**	**Exergy Destruction (Percentage b)**
Generator	1.926	55.18	0..9915
Solution heat exchanger	0.002395	0.000686	0.0012
Absorber	1.414	40.51	0.7279
Evaporator	0.0894	2.561	0.046
Condenser	0.0613	1.75	0.0316
Total	3.49	100	1.7982

From the results, it can be concluded that exergy destruction rate is higher in the combustion chamber and the generator in case of gas turbine system and vapour absorption refrigeration system. All the principal irreversibilities like chemical reaction, heat transfer, and friction are present in combustion chamber. The total exergy loss per thermodynamic cycle increases with the addition of IAC feature because more reaction process (more components) will be implemented into the system. Again the exergy destruction in VAR system matches reasonably well with the value of ref **[11]**.The exergy destruction in VAR system is negligible in comparison to combustion chamber.

4.2.1.4 Comparison between Simple Cycle, STIG Cycle and IAC & Process Steam Cycle

Table 4.20 below shows the various sytem parameters and exergy destruction in system components.

Table 4.20 Power output, efficiency and exergy destruction data for simple cycle, STIG cycle and IAC & process steam cycle

System	**Existing Simple Cycle**		**STIG Cycle**		**IAC and Process Steam Cycle**	
Power output (MW)	50.912	(52.14)	87.797	(91.85)	55.761	(62.79)
Power generation efficiency (%)	27.02	(29)	38.78	(39.9)	27.62	(32.0)
First law efficiency (%)	84.61		38.78		53.25	
Second law efficiency (%)	75.98		37.06		51.08	
Power-to-heat ratio	0.4371		-		-	
Fuel-air ratio	0.0298		0.03626		0.0342	
Steam injection ratio	-		0.2397		-	
Compressor work (MW)	68.916		68.916		67.656	
Exergy Destruction Data(MW) **Component**						
1. Compressor	4.926	(5.1)	4.926		4.923	(6.49)
2. Combustor	70.03	(69.38)	104.039		71.466	(75.34)
3. Turbine	6.72	(7.16)	8.184		6.687	(6.52)
4. HRSG	14.618		19.674		15.062	(15.95)
5. VARS system	-	-	-		3.49	(3.27)
6. Total exergy loss	96.294		136.823		101.628	

The calculated power output is 50912 kW and the efficiency is 27.02% which are very close to the ref **[11]** data (at 52140 kW and 29%). Similar is the case for system with steam injection and inlet air cooling & process steam cycle. Since the compression work= $-\int vdP$, where v is the specific volume of inlet air, when the inlet air is cooled down, the compression work is thus reduces. At the same compression volume, the mass flow rate per compression is inceased, therefore, the output is also increased. The minor difference is due to assumption made in the value of some input parameters.

The exergy destruction data summarized in Table 4.20 have been drawn in Figures from 4.3 to 4.8.

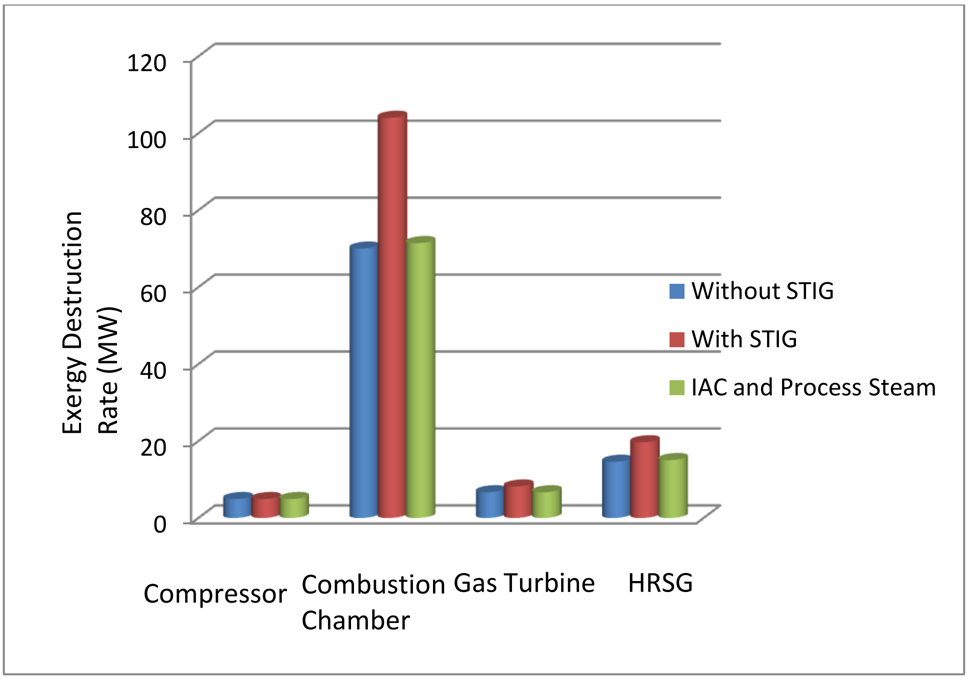

Fig. 4.3 Exergy destruction rate in system components

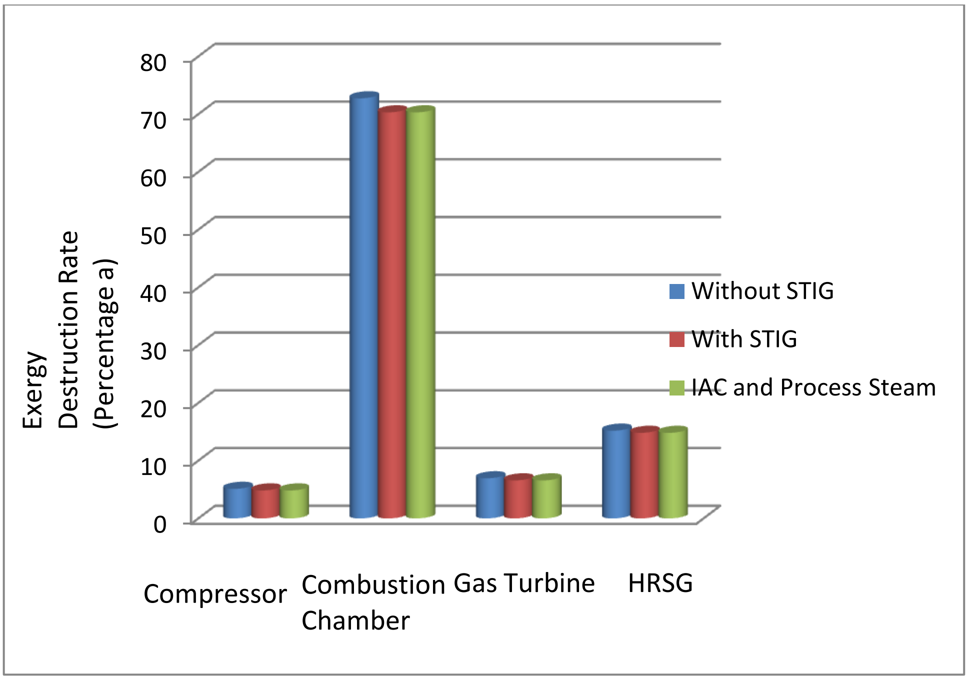

Fig. 4.4 Exergy destruction rate in system components as a percentage of total exergy destruction

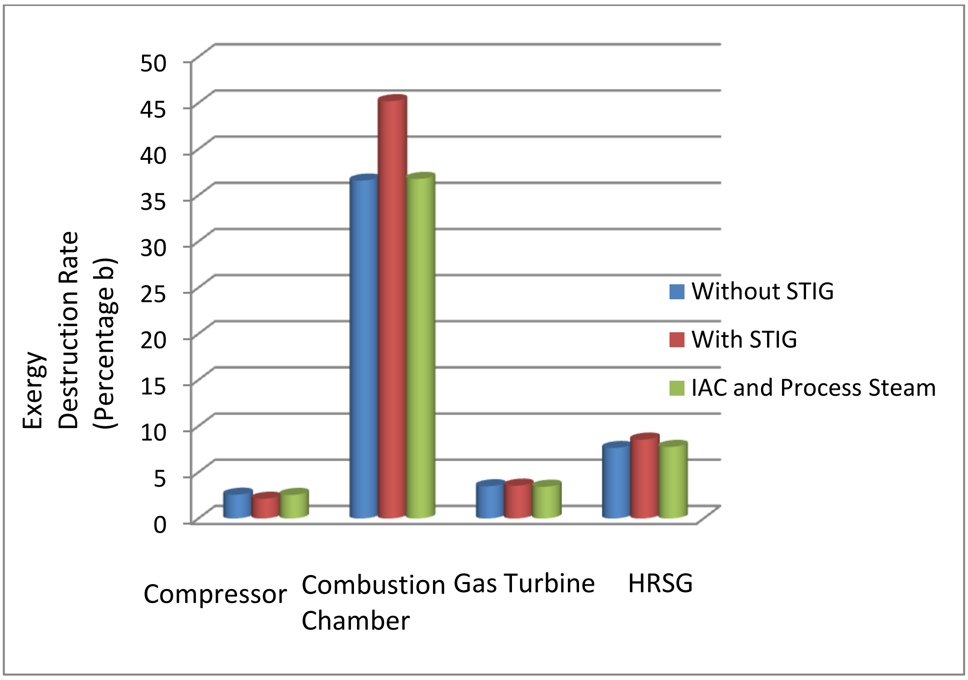

Fig. 4.5 Exergy destruction rate in system components as a percentage of fuel exergy supplied to the system

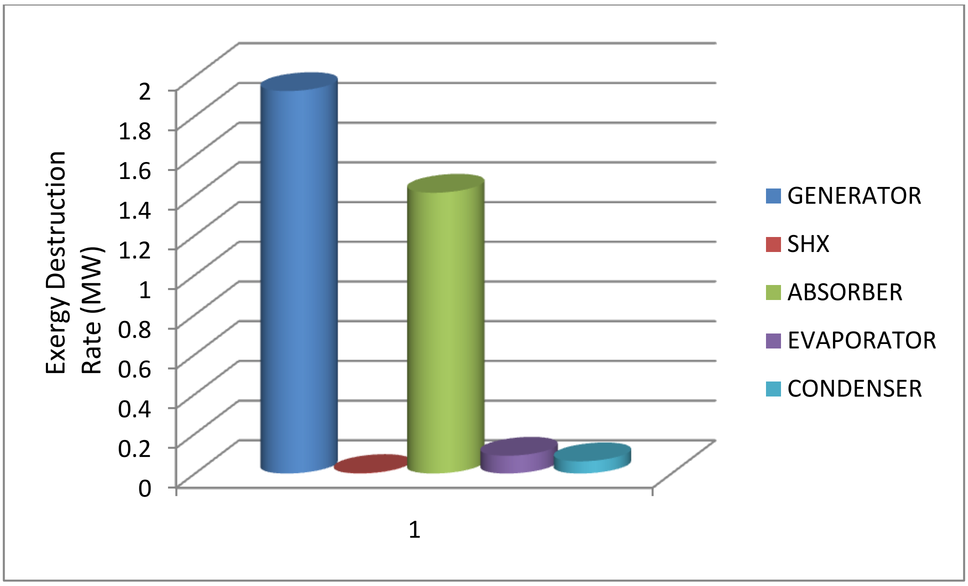

Fig. 4.6 Exergy destruction rate in system components of VAR system

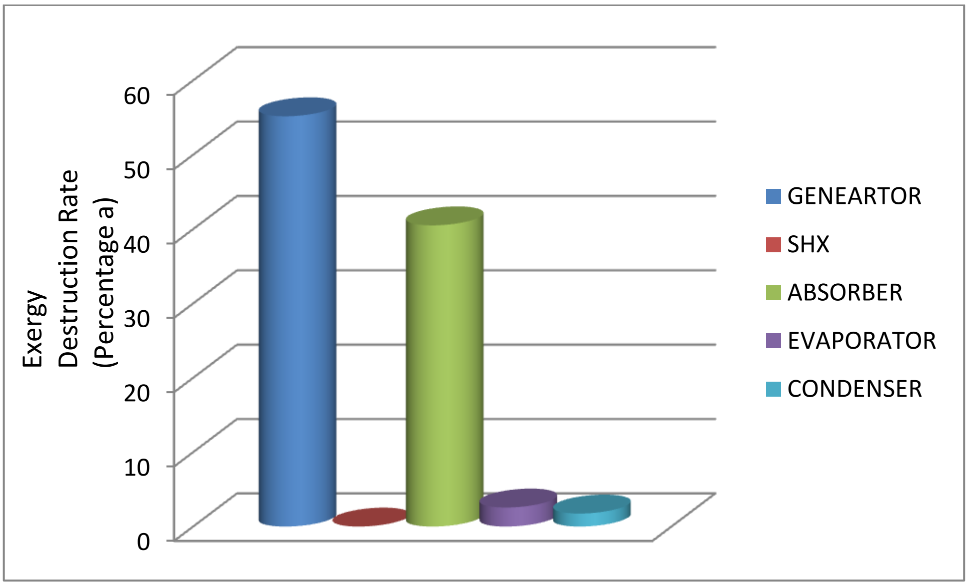

Fig. 4.7 Exergy destruction rate in system components of VAR system as a percentage of total exergy destruction

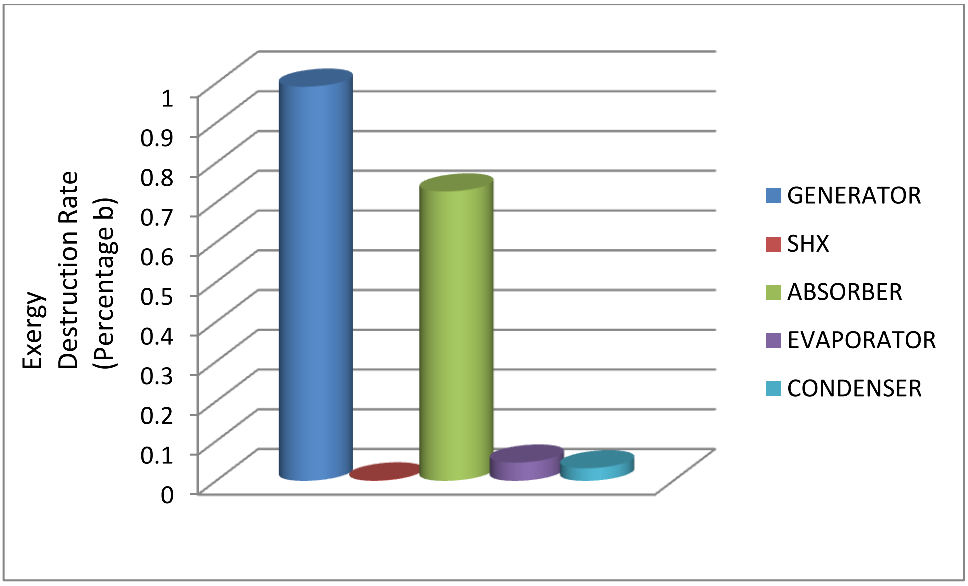

Fig. 4.8 Exergy destruction rate in system components of VAR system as a percentage of fuel exergy supplied to the system

The exergy destruction rate represents the waste of energy availabilities. Exergy destruction of all the components have been calculated to enhance the understanding of the cycle performance. Data clearly identify the combustion chamber as the major site of thermodynamic inefficiency. The nest prominent site is the heat-recovery steam generator (HRSG). Small contribution to inefficiency is made by the gas turbine and air compressor.

The exergy destructions in these componenents stem from one or more of three principal irreversibilities associated, respectively, with chemical reaction, heat transfer, and friction. All these irreversibilities are present in the combustion chamber, where chemical reaction is the most significant source of exergy destruction. For the HRSG, the friction and heat transfer are the sources of exergy destruction, with the most significant irreversibility related to the stream-to-stream heat transfer. Exergy destruction in the adiabatic gas turbine and air compressor is caused primarily by friction. Steam injection will also increase the exergy destruction due to mixing in the combustor. The total exergy loss per thermodynamic cycle increases with the addition of STIG feature because more reaction process will be implemented into the system. However, the modified system has a substantial gain in power output and efficiency.

The benefits of adding the STIG features can be estimated from Figure 4.9 in which both the power output and generation efficiency are increased with the increase of injection ratio.

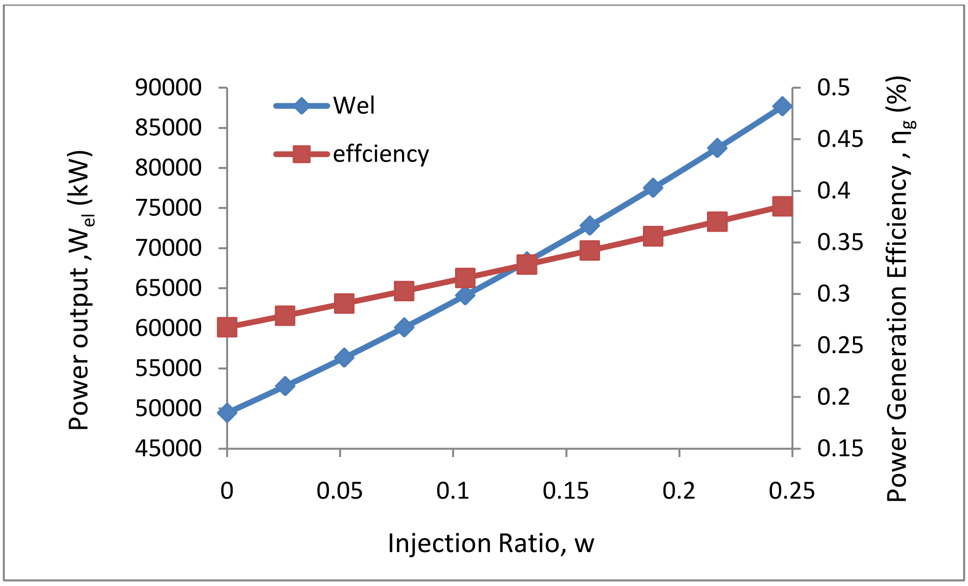

Fig. 4.9 Effect of injection ratio on power output and generation efficiency

The profound effect from STIG alone is because the required pressure of injected steam is obtained from a pump. Since the pumping work is 2-3 orders of magnitude smaller than that of compressor, the net power output produced by steam is, thus, much higher than that of air per unit mass flow rate. In addition, the specific heat of superheated steam is almost double the value of air and the enthalpy of steam is higher than that of air at a certain temperature. Therefore, the STIG method is a very effective way to boost the net power output and increase the overall efficiency of gas turbine.

Although the amount of steam required by the IAC and process steam is only a fraction of the amount that can be generated from waste energy, the allowable amount of steam that can be injected into the combustor for the STIG feature is more than the maximum remaining amount that can be generated from waste energy.

From the result of Table 4.20, we can discern that the effect of STIG is more profound than that of the IAC in improving the power output and efficiency.

The maximum amount of injection steam is limited by the available energy recovered from the HRSG. The maximum steam injection ratio for the frame 7B GENSET is estimated to be around 0.2397 The maximum injection at 0.2397 is still below the allowable injection limit for the frame 7B GENSET. From Figure 4.17, it is found that the power output can be substantially increased from 50912 to 87670 kW, and the associated efficiency can be increased from 27.02% to 38.49% during a typical local summer condition.

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4.2.2 Parametric Studies

The compressor air inlet temperature is the continuously varying parameter in power plants. The effect of varying the compressor air inlet temperature has been studied on the first law efficiency (η_I), second law efficiency (η_{II}) ,power-to-heat ratio (R_{PH}), fuel-air ratio and steam injection ratio in three different cases discussed earlier. The first law efficiency and power-to-heat ratio is obtained by the energy balance approach or the first law analysis of the cycle. Also, exergy destruction in system components has been studied with variation in inlet temperature of air. However, the exergy destruction or thermodynamic losses of each component and second law efficiency (η_{II}) of the cycle has been investigated under the exergy balance approach or second-law analysis of the cycle. To study the effect of compressor air inlet temperature on the performance parameter of the system, operating under different conditions, the following common characteristics and their corresponding ranges were chosen-

1. **Compressor air inlet temperature (T_1) -** The air reference temperature of the compressor has been varied for a range of 10 to 50˚C.
2. **Turbine Inlet temperature (TIT) and pressure ratio –** The turbine inlet temperature and pressure ratio of the compressor has been taken as 1264 K and 9:1 respectively.
3. **Characteristics of fuel-** Natural gas (methane) is taken as fuel having the lower heating value of 802361 kJ/kmol.
4. **Steam pressure-** The high pressure and low pressure of steam has been varied in the range from 900 to 1400 kPa and 400 to 800 kPa respectively.
5. **System component losses-**
   Pressure drop in compressor = 1%
   Pressure drop in combustion chamber = 5%
   Pressure drop in gas turbine = 1%
   Pressure loss in HRSG = 5%

6. **System component efficiencies**: The system component efficiencies are given below

   Compressor isentropic efficiency η_{Sc} = 86%
   Gas turbine isentropic efficiency η_{ST} = 87%
   Efficiency of combustion chamber η_{cc} =98%
   Electric generator efficiency η_g = 98%
   Effectiveness of generator = 0.8
   Effectiveness of solution heat exchanger = 0.9
   Effectiveness of air heat exchanger = 0.75
   Efficiency of solution pump = 0.9

4.2.2.1 Effect of variation of inlet air temperature on first law and second law efficiency-

Figures 4.10 and 4.11 shows the variation of first law efficiency and second law efficiency for the simple cycle, STIG cycle and IAC & process steam (trigeneration) cycle. As air temperature rises, its density falls. Thus, mass flow rate is reduced but the volumetric flow rate remains constant. Again power output is proportional to mass flow rate of air passing through the compressor therefore net power output is reduced as compressor air intake temperature rises. Since first law efficiency is the ratio of net work to the heat supplied by the fuel therefore first law efficiency decreases as inlet temperature increases. The value of k i.e. fraction of steam injected into the combustion chamber is taken equal to zero. The variation in first law efficiency is almost similar for the case of STIG cycle and IAC cycle but the first law efficiency is higher in case of STIG cycle. Steam injection in the combustion chamber increases the net power output therefore it is higher. By cooling the inlet air, the compressor work reduces due to increase in density. But volumetric flow rate remains constant and mass flow rate of air passing through the compressor reduces considerably and compressor work decreases. Again increasing compressor air inlet temperature reduces second law efficiency. In case of simple cycle, all the energy of exhaust flue gases is utilized in generating steam. The process heat available in the form of steam is added in the net output therefore first law efficiency is higher in case of simple cycle. The variation of second law efficiency is also similar for the STIG and trigeneration cycle. In case of second law efficiency, exergy comes into picture in place of energy. The same concept for analysis can be applied in case of second law efficiency.

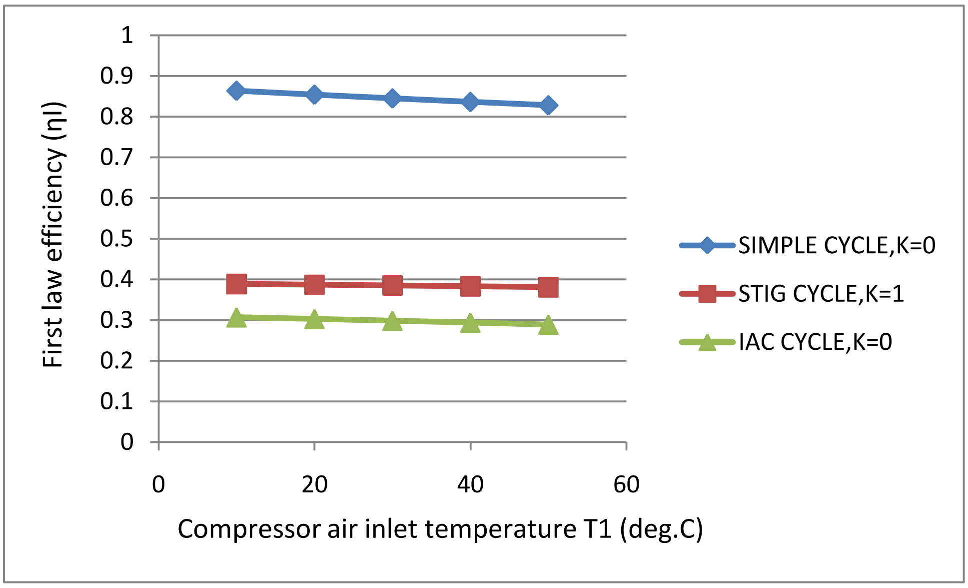

Fig.4.10 Effect of variation of compressor air inlet temperature on first law efficiency

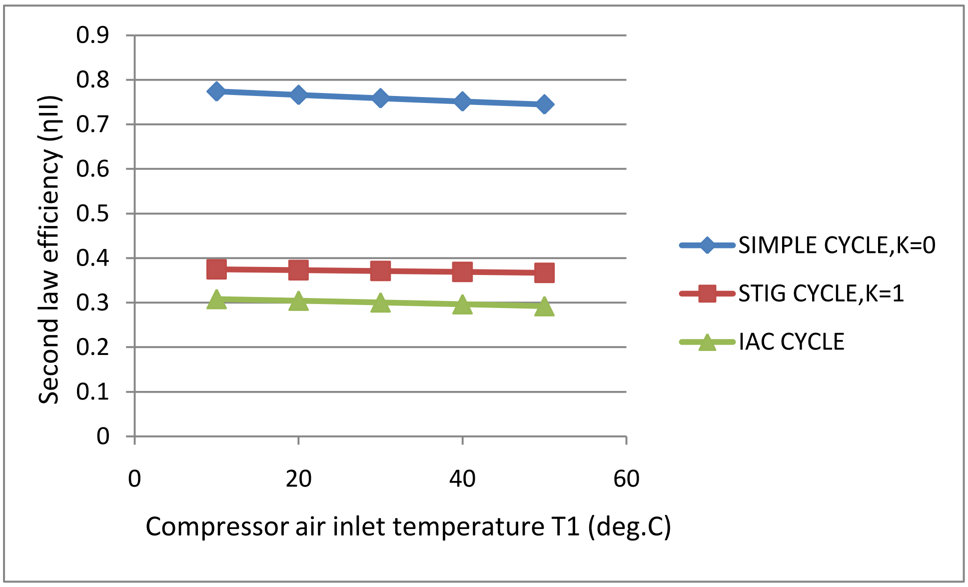

Fig.4.11 Effect of variation of compressor air inlet temperature second law efficiency

4.2.2.2 Effect of variation of inlet air temperature on power-to-heat ratio, fuel air ratio and injection ratio-

Figures 4.12, 4.13 and 4.14 shows the variation of power-to-heat ratio, fuel-air ratio and steam injection ratio for the simple cycle, STIG cycle and IAC & process steam (trigeneration) cycle. Power-to-heat ratio has not been drawn for the case of STIG cycle because in this case all the generated superheated steam is utilized for injection purpose therefore no process heat in the form of steam is available. From the Figure 4.12, it is clear that as inlet temperature rises, power-to-heat ratio reduces. It is due to the fact that increasing inlet temperature of air increases compressor work and net work decreases. In case of system with inlet air cooling and process steam, power heat ratio decreases at slow rate. In this case, the generated saturated steam is used for cooling the air and remaining process steam is used for process heating purposes, therefore process heat available decreases slightly but cooling inlet air reduces compressor work and net output increases. Again the increase in net power is less as compared to decreases in process heat therefore power-to-heat ratio slightly decreases. Figure 4.13 shows the variation of fuel-air ratio. Increasing inlet temperature increases the temperature of air at inlet to combustion chamber therefore less amount of fuel is required for burning the fuel in case of simple cycle. In case of STIG cycle, the fuel-air ratio is considerable higher because in case of steam injection, the combustion temperature is maintained at constant temperature. To maintain the combustion chamber at constant temperature, excess fuel quantity is added in the combustion chamber is somewhat more. Fuel-air ratio variation is nearly similar for the case of simple cycle and IAC & process steam cycle. The steam injection ratio decreases as inlet temperature increases because higher inlet temperature causes in generation more amount of gases and therefore steam injection ratio decreases as shown in Figure 4.14.

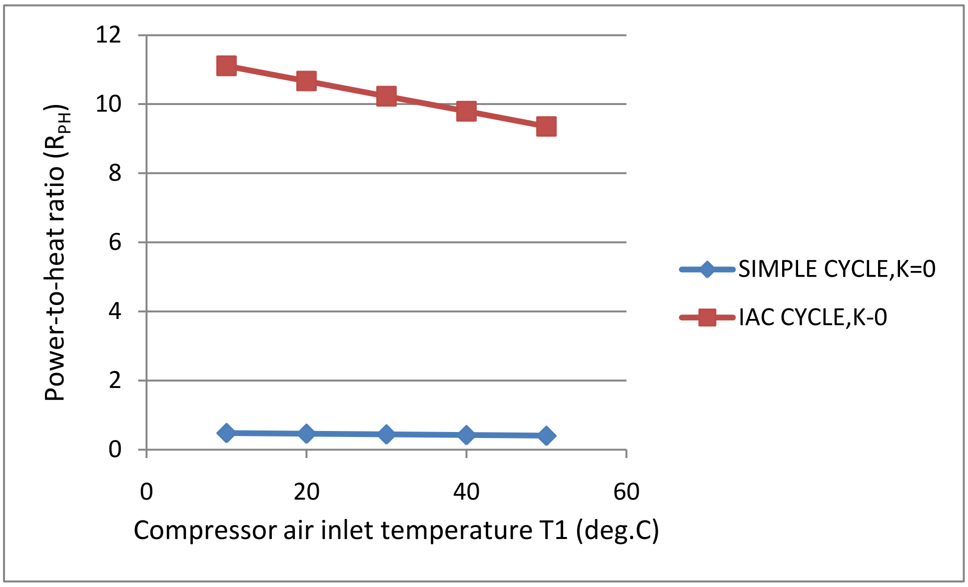

Fig.4.12 Effect of variation of compressor air inlet temperature on power-to-heat ratio

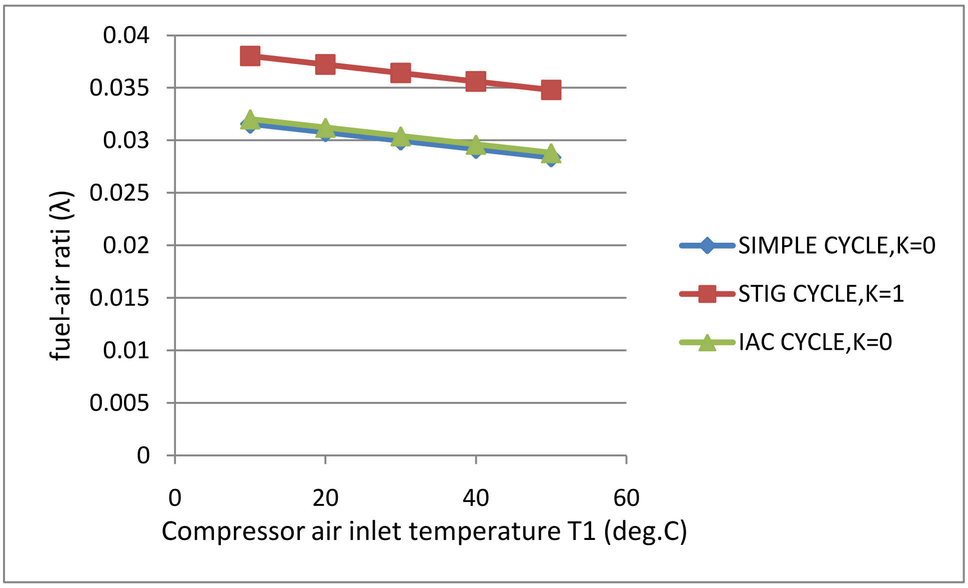

Fig.4.13 Effect of variation of compressor air inlet temperature on fuel-air ratio

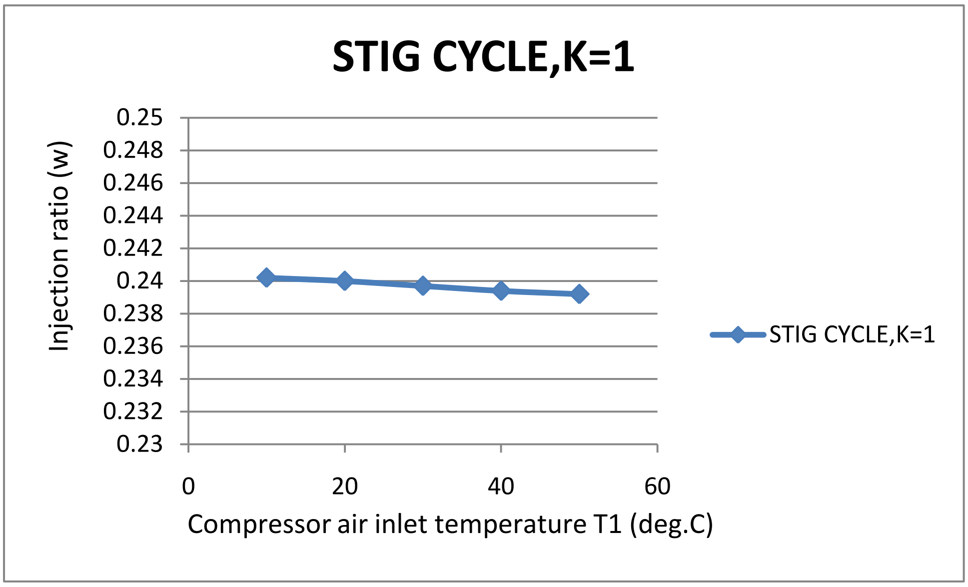

Fig.4.14 Effect of variation of compressor air inlet temperature on steam injection ratio

4.2.2.3 Effect of variation of high pressure and low pressure of steam generation on first law and second law efficiency, power-to-heat ratio, fuel-air ratio and steam injection ratio –

By varying the high pressure and low pressure of steam generation, different performance parameters has been plotted for simple and STIG cycle which is shown in Figure 4.15 and 4.16. Here saturated steam is generated at low pressure which can be used for process heating in plants and superheated steam is generated at high pressure which is used for steam injection in the combustion chamber. Effect of varying the low pressure has no effect on fuel-air ratio because steam has no role to play in simple cycle. It is used externally as process steam. As the low pressure increases, the power-to-heat ratio increases continuously. Because as low pressure increases, less amount of steam will be available and process heat available in the form of process steam decreases and power-to-heat ratio increases. First and second law efficiency decreases by increasing the low pressure of steam due to reduced quantity of process heat available but first law efficiency is more than second law efficiency. In case of cogeneration system with steam injection, the fuel-air ratio and steam injection ratio decreases. The same concept applies here also.

By increasing pressure, amount of steam generated decreases but at the same time its energy increases. Since steam injection ratio is the mass of steam injected to the mass of gases formed therefore it decreases slightly with increase in high pressure of steam. Similarly, at high pressure of steam, its temperature would be higher therefore fuel-air ratio decreases. The variation is same for first law efficiency and second law efficiency for the STIG cycle. The first and second law efficiency is less in case of STIG cycle as compared cogeneration system without steam injection because in this case, all the process steam is totally utilized for injecting into the combustion chamber for enhancement of power.

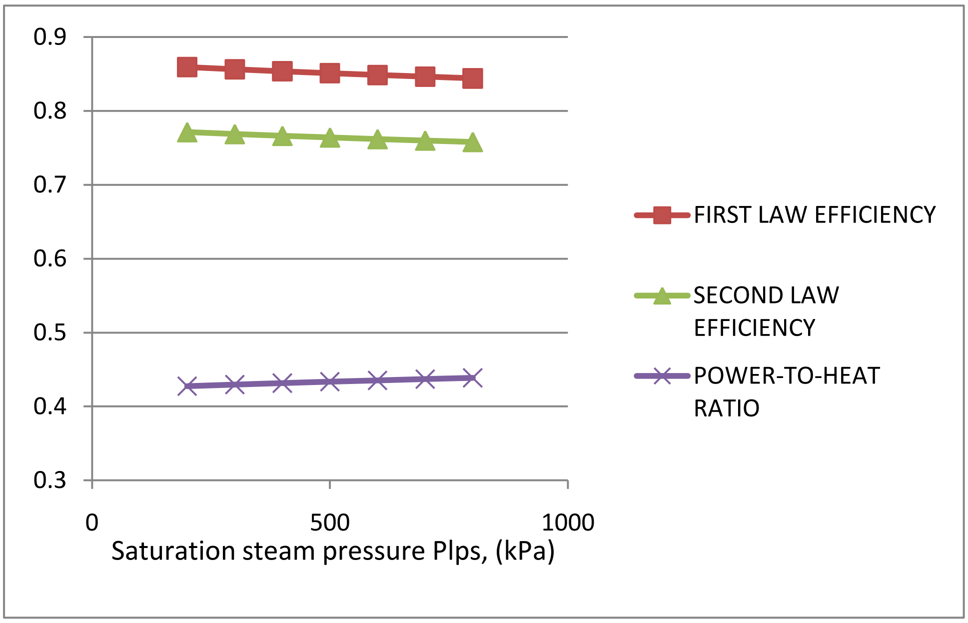

Fig.4.15 Effect of variation of low pressure of steam on fuel-air ratio, first and second law efficiency and power-to-heat ratio

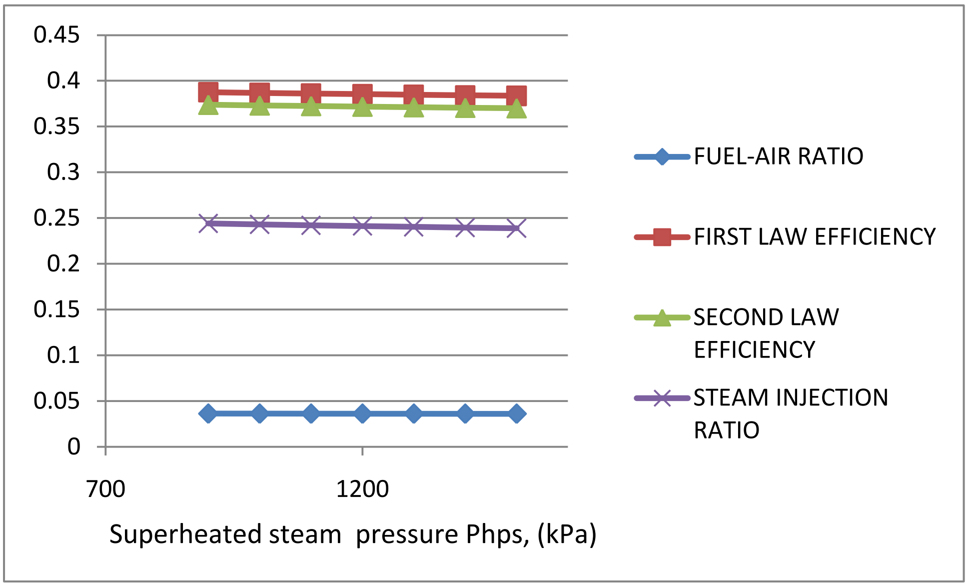

Fig.4.16 Effect of variation of high pressure of steam on fuel-air ratio, first and second law efficiency and power-to-heat ratio

4.2.2.4 Effect of variation of compressor air inlet temperature on exergy destruction in system components–

Figures from 4.17 to 4.18 below shows the variation of exergy destruction in system components with varying air intake temperature for simple cycle, STIG cycle and IAC & process steam cycle. Figures clearly identify that the combustion chamber is the major site of thermodynamic inefficiency. The next prominent site is the heat-recovery steam generator (HRSG). Small contribution to inefficiency is made by the gas turbine and air compressor.

In case of compressor, increasing inlet air temperature increases exergy destruction. The exergy destruction in compressor is almost similar for the three cycles. In case of cogeneration with and without steam injection, the exergy destruction is same for the compressor therefore exergy destruction lines coincides as shown in Figure 4.17. But in case of inlet air cooling, the exergy destruction is slightly lower. This is because of reduction in inlet air temperature due to inlet air cooling. In case of gas turbine, the exergy destruction is highest in case of steam injection and lowest in case of simple cycle. As far as HRSG is concerned, its value is again higher in case of steam injection and lowest in trigeneration cycle. In case of steam injection, the steam added is a mixing process which results in increasing value of exergy destruction. By employing inlet air cooling, the exergy destruction in HRSG can be reduced which ultimately lead to reduce the exergy losses through the stack as seen in Figure 4.19 The exergy loss through the stack will not only waste the available energy but also reflect the degree of thermal pollution to our living environment. The exergy destruction in combustion chamber is highest among all the system componenents due to more temperature existing in combustion chamber. All the irreversibilities like chemical reaction, heat transfer and friction are present in the combustion chamber, where chemical reaction is the most significant source of exergy destruction. Also from the Figure 4.20 it can be concluded that steam injection increases the exergy destruction in the combustion chamber drastically due to mixing in the combustor.

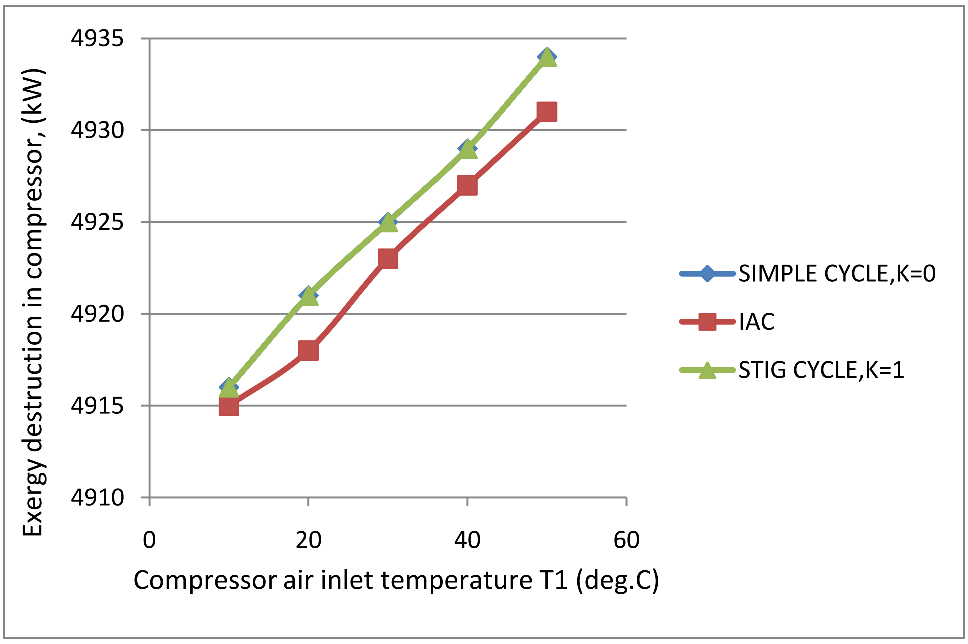

Fig.4.17 Effect of variation of compressor air inlet temperature on exergy destruction in compressor

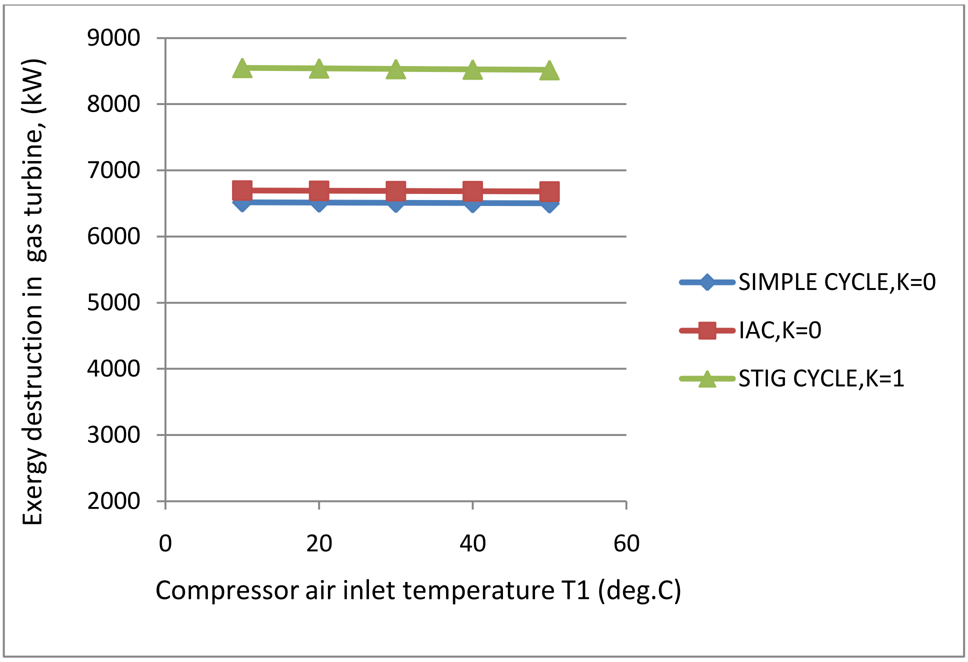

Fig.4.18 Effect of variation of compressor air inlet temperature on exergy destruction in gas turbine

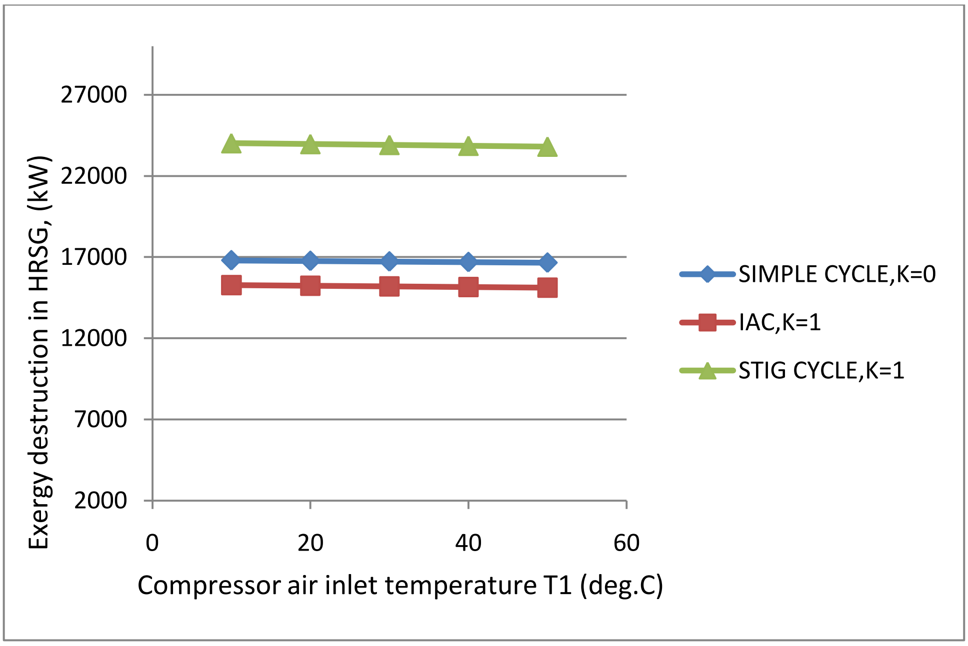

Fig.4.19 Effect of variation of compressor air inlet temperature on exergy destruction in HRSG

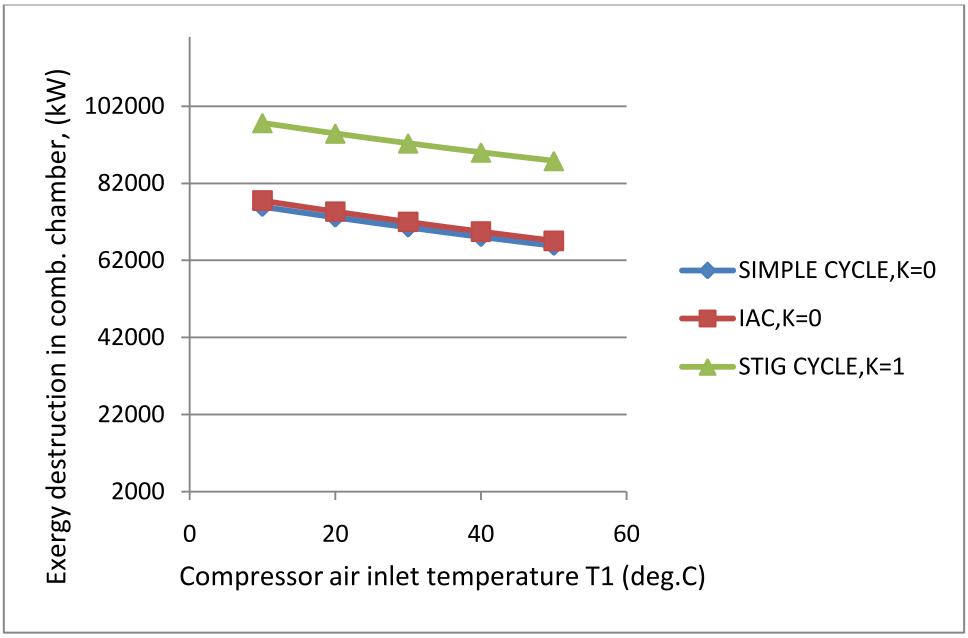

Fig.4.20 Effect of variation of compressor air inlet temperature on exergy destruction in combustion chamber

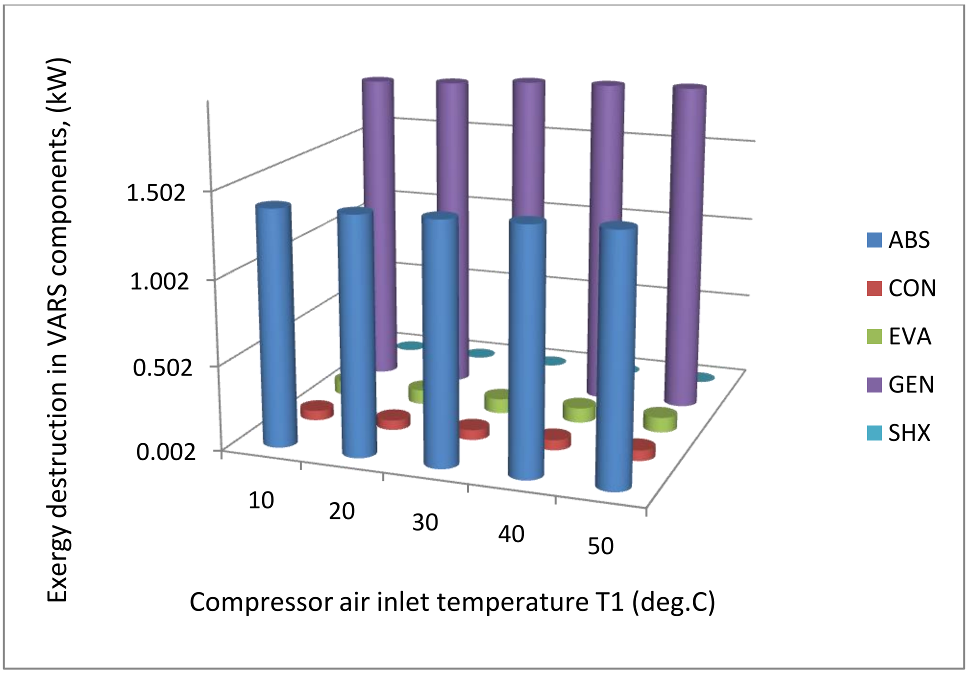

Fig. 4.21 Exergy destruction rate in components of VAR system with variation in inlet temperature of air

CHAPTER 5

CONCLUSION AND FUTURE SCOPE OF WORK

The reasons for having a low efficiency and a high exhausted temperature can be discerned from the fundamental thermodynamic analysis, and the methods used in the past to improve the efficiency were focussed on either increasing the expansion work or decreasing the compression work. Among the many performance-improvement techniques, the steam-injection gas turbine (STIG) and inlet-air cooling (IAC) are well suited for a retrofitting project without destroying to its original integrity. Steam injection and inlet air cooling are well-proven technologies that can effectively improve power output and power generation efficiency for a simple cycle gas turbine GENSET. In this study, an existing Fame 7B simple cycle GENSET was considered as the basic system and converted into modified system with either the IAC or/and STIG features. The steam needed in the STIG and IAC features is generated from the energy recovered from the system's own exhaust gases, additional environment benefits can also be realized. Under the average local summer weather, the benefit of adding the STIG feature can substantially improve the power output from 50.912 MW to 87.670 MW, and the associated efficiency can be increased from 27.02% to 38.49%.

In examining the exergy destruction rate for all the components, the combustor has the largest exergy destruction and shows the major site of thermodynamic inefficiency because of the large irreversibilities arising from the chemical reaction and heat transfer. Steam injection will also increase the exergy destruction due to mixing in the combustor. The total exergy loss per thermodynamic cycle increases with the addition of the IAC or STIG features because more componenets (more reaction process) will be implemented into the system. However, the modified system has a substantial gain in power output.

The present study as presented a methodology based on first and second law for the thermodynamic analysis of gas turbine cogeneration system integrated with IAC and STIG features. This methodology is relatively simple and systematic. The most interesting aspect of the study is that thermodynamic analysis of gas turbine cogeneration based on first-law alone is inadequate and more meaningful evaluation must include second-law analysis. The cogeneration system is more efficient by incorporating features of IAC and STIG as compared to simple cycle gas turbine.

Recommendations for future work

The present work can be further extended in following directions-

1. The present cogeneration system can be extended by incorporating the concept of reheat and regenerative steam-injection gas-turbine systems.
2. Absorption unit and an intercooling unit, ejection cooling system, fogging system or double effect vapour absorption refrigeration system can be employed to cool the air of the compression system in place of single effect vapour absorption refrigeration system.
3. Presently the STIG technology is applied only to the gas turbines cycle. It can be extended to the combined power cycle which are now well established.
4. The present study can be further extended for Techno-economic analysis.

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APPENDIX I

Terminology Used in Cogeneration System-

Energy- A measure of the amount of 'work' that can be done by, or is needed to operate, an energy conversion system, sometimes measured in 'joules' (J) or 'kilowatt hours' (kWh).

Enthalpy- It is a property and is defined as the sum of the internal energy U and the PV product. It has been taken from the Greek word enthalpien, which means to heat.

Enthalpy of Combustion (h_{RP}) - It is the difference between the enthalpy of the products and the enthalpy of the reactants, each on per mole of fuel basis, when the combustion occurs and both reactants and products are at the same temperature and pressure. For hydrocarbon fuels the enthalpy of combustion is negative in value since chemical internal energy is liberated in the reaction.

Heating Value- The heating value of a fuel is a positive number equal to the magnitude of the enthalpy of combustion. Two heating values are recognized by name, the higher heating value (HHV) and the lower heating value (LHV). The higher heating value exceeds the lower heating value by the energy that would be required to vaporize the liquid water formed at the specified temperature. The values of HHV and LHV are typically reported at a temperature of 25˚C and a pressure of 1 bar. These values also depend on whether the fuel is a liquid or a gas.

Exergy- It is the maximum theoretical useful work (shaft work or electrical work) obtainable as the systems interact to equilibrium, heat transfer occurring with the environment only. Exergy is measure of the departure of the state of the system from that of the environment. It is therefore an attribute of the system and environment together. Exergy can be regarded as an extensive property of the system.

In the absence of nuclear, magnetic, electrical, and surface tension effects, the total exergy of a system E can be divided into four components: physical exergy E^{PH}, kinetic exergy E^{KN}, potential exergy E^{PT} and chemical exergy E^{CH}.

$$E = E^{PH} + E^{KN} + E^{PT} + E^{CH}$$

Exergy Destruction- It is commonly referred to as the availability destruction, the irreversibility, and the lost work.

Physical Exergy- The physical exergy is the maximum theoretical useful work obtainable as the system passes from its initial state where the temperature is T and the pressure is P to the restricted dead state where the temperature is T_o and the pressure is P_o.

It results from the temperature and pressure of the substance measured with respect to the temperature and pressure of the environment. The physical exergy is used for analysis of physical processes, in which, the kind of the considered substance is unchanged.

Chemical Exergy- It is the exergy component associated with the departure of the chemical composition of a system from that of the environment. The chemical exergy is the maximum theoretical useful work obtainable as the system passes from the restricted dead state to the dead state.

APPENDIX II

Table 7.1- Standard Molar Chemical Exergy e^{-CH} (kJ/kmol), of Various Substances at 298.15 K and P_o (1.013 Bar)

Substance	Formula	Model ^{A}	Model ^{B}
Nitrogen	N_2(g)	639	720
Oxygen	O_2(g)	3951	3970
Carbon dioxide	CO_2(g)	14176	19870
Water	H_2O(g)	8636	9500
Water	H_2O(L)	45	900
Hydrogen	H_2(g)	235249	236100
Methane	CH_4(g)	824348	831650

APPENDIX III

Equation of Entropy Used in Calculating Physical Exergy in Components of Vapour Absorption Refrigeration System

$$\begin{aligned} S = [&(5.127558 \times (10^{(-1)})) - (1.393954 \times (10^{(-2)}) \times X) + (2.924145 \\ &\times (10^{(-5)}) \times (X^2)) + (9.036 \times (10^{(-7)}) \times (X)^3) + (1.22678 \\ &\times (10^{(-2)} \times T) - (9.15682 \times (10^{(-5)}) \times X \times T) + (1.820453 \\ &\times (10^{(-8)}) \times (X^2) \times T) - (7.991806 \times (10^{(-10)}) \times (X^3) \times T) \\ &+ (-1.364 \times (10^{(-5)} \times (T^{(2)}) + (1.068 \times (10^{(-7)}) \times X \times T^{(2)}) \\ &- (1.381109 \times (10^{(-9)}) \times X^2 \times T^2) + (1.529784 \times (10^{(-11)}) \times X^3 \\ &\times T^2) + (1.021501 \times (10^{(-8)}) \times T^3)] \end{aligned}$$