THERMODYNAMIC ANALYSIS OF GAS TURBINE BASED TRIGENERATION SYSTEM

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By

MANOJ KUMAR
Roll no. 12297

Under the Supervision of

RAJ KUMAR SINGH **Dr. S.S.KACHHWAHA**

Department of Mechanical Engineering,
Delhi College of Engineering, University of Delhi
DELHI -110042
INDIA
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ABSTRACT

Trigeneration is the simultaneous production of power/electricity, hot water and/or steam, and chilled water from one fuel. Basically, a trigeneration power plant is a cogeneration power plant that has added absorption chillers for producing chilled water from the heat that would have been wasted from a cogeneration power plant. Trigeneration system efficiency is higher than both simple and cogeneration system.

In this present research work case studies of cogeneration (full load and part load) and trigeneration plants have been performed. For this purpose operating parameter (mass flow rate, Pressure, Temperature etc) at inlet and outlet of each system component have been determined.

By using EES calculated parameter (Enthalpy, Entropy, Exergy etc) has been determined, With the help of these parameters, System performance parameters (first law, second law efficiency, exergy) have been estimated.

Parametric study has been done using combine first and second law approach to investigate the effects of overall pressure ratio, turbine inlet temperature, inlet air temperature on exergy destruction, first law efficiency, electrical to thermal energy ratio and second law efficiency of the components and overall system. Thermodynamic analysis shows the exergy destruction in combustion chamber and HRSG is significantly affected by the pressure ratio and turbine inlet temperature. It is also indicate the maximum exergy is destroyed during the combustion and steam generation process which represents 82% of total exergy destruction in the overall system.

Keywords: Cogeneration system, Trigeneration system, Absorption system, Exergy, Air cooler, Heat recovery

CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

E	Exergy (MW)
E^{CH}	Specific chemical exergy (kJ/kmol)
E^{PH}	Specific physical exergy (kJ/kmole)
f_c	Circulation rate
$\dot{h}$	Specific enthalpy of the fluid (kJ/mole)
M	Molecular weight (kg/kmol)
m	Mass flow rate (kg/s)
n	Molar flow rate (kmol/s)
P	Pressure (bar)
PP	Pinch point (˚C)
Q	Heat transfer (kJ)
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)
x	Molar fraction
X	Concentration of LiBr
y	Exergy destruction ratio

Subscripts

a	Air
A	Absorber
COMP	Compressor
CC	Combustion Chamber
C	Condenser
fw	Feed water

cv	Control volume
D	Destruction
ECO	Economizer
E	Evaporator
f	Fuel
G	Generator
HRSG	Heat Recovery Steam Generator
k	k^{th} component
L	loss
o	reference
p	Product or gases
P	Pinch point
r	Refrigerant
S	Isentropic
SHX	Solution Heat Exchanger
REG	Regenerator
sat	Saturated
sup	Superheated
st	Steam
T	Turbine
w	Water

Superscript

CH	Chemical
KN	Kinetic
PH	Physical
PT	Potential
T	Total

Greek symbol

η	Efficiency
λ	Fuel-Air ratio

Acronyms and abbreviations

VAR	Vapor Absorption Refrigeration System
COP	Coefficient of Performance
EES	Engineering Equation Solver
GT	Gas Turbine
HHV	Higher Heating Value
IAC	Inlet Air Cooling
LHV	Lower Heating Value
AHE	Air Heat Exchanger

Chapter 1

INTRODUCTION

1.1Definition and Historical Development

Trigeneration is the simultaneous production of power/electricity, hot water and/or steam, and chilled water from one fuel. Basically, a trigeneration power plant is a cogeneration power plant that has added absorption chillers for producing chilled water from the heat that would have been wasted from a cogeneration power plant. Trigeneration system efficiency is higher than both simple and cogeneration system.

During the operation of conventional power plant, large quantity of heat is rejected in atmosphere (steam condensers, cooling towers etc.) or with the exhaust gases. Most of this heat can be recover and used to cover thermal needs thus increasing the efficiency from 30%-40% of power plant to 80% -90% of a cogeneration and tri-generation system. A comparison between cogeneration and separate production of heat and electricity from the point of efficiency is shown in Figure1.1 based on their typical values of efficiencies.

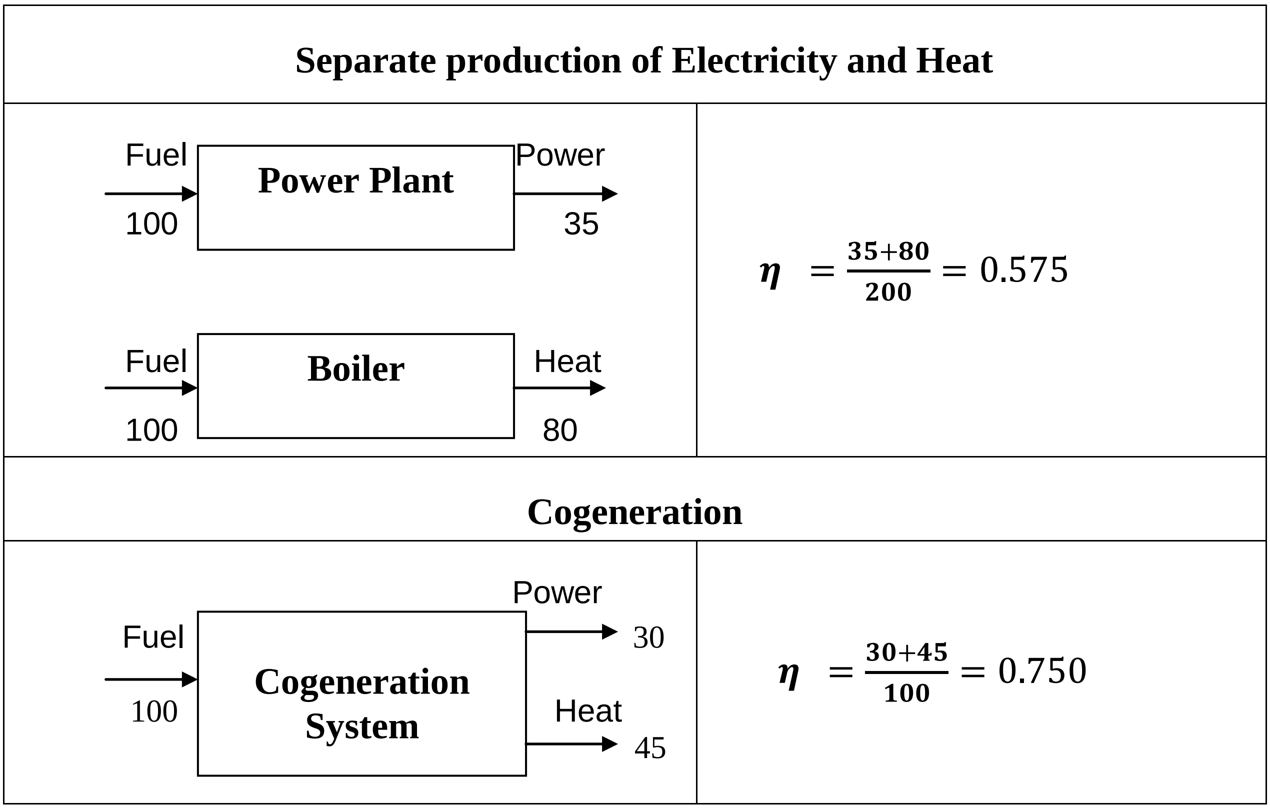

Fig.1.1 Efficiency comparison of cogeneration and separate production of power and heat

Cogeneration first appeared in late 1880 in Europe and U.S.A. during the early part of 20th century most industrial plants generate their own electricity using coal or gas fired boiler and steam turbine generators. Many of the plant used their exhaust steam for industrial processes. It has been estimated that as much as 58% of total power produced in the early 1900's by on-site industrial power plants in U.S.A. was cogenerated.

The aforementioned trend in cogeneration started being invented after the first dramatic rise of fossil fuel costs in 1973. Systems that are efficient and can utilize alternative fuels have become more important in the face of price rises and uncertainty of fuel supplies. 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. Methods of stimulating the use of cogeneration are seen in three major forms regulations, monetary incentives and financial support of research and development. Research, development and demonstration projects realized during the last 25 years led to a significant improvement of the technology, which now is mature and reliable.

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 through Trigeneration in core industries such as sugar and rice mills, distilleries, petrochemical sector and industries such as fertilizers, steel, chemical, cement, pulp and paper, and aluminium.

An attempt has been made to implement a suitable Trigeneration 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 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.

1.3 Motivation of present research work

Thermodynamic analysis can be perfect tool for identifying the ways for improving the efficiency of fuel use, and determining the best configuration and equipment size for a Trigeneration plant. 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. 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 that motivate for doing this research work.

1.4 Organization of the Report

Chapter 1 expresses the introduction of Trigeneration, its history and development, review of gas turbine Trigeneration system and its importance. The brief explanation has been showed about features and applications of gas turbine.

Chapter 2 shows literature review section in which various researches related to gas turbine based trigeneration system. Also the conclusion of the literature survey study has been expressed in this chapter.

Chapter 3 describes formulation of Trigeneration system in which standard cogeneration system integrated with VAR 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, case studies (part and full load) and parametric study for trigeneration.

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

Chapter 2

LITERATURE REVIEW

The focus of this review is on Trigeneration specific application. Trigeneration 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 project is enlisted below.

2.1 Summary of Literature Review

Major area of research in Gas Turbine Trigeneration included the research on cogeneration plant and Refrigeration system (Vapour Absorption refrigeration system) to improve the performance thermodynamically with reduced expenditure.

Moran **[1]** has established a design methodology for the gas turbine cogeneration system. This simple system is integrated with Regenerator and a 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 and energy destruction at different points are also tabulated. The waste heat from turbine (exhaust gases) has been used to produce steam for other processes.

Jolly et al **[2]** analysed Gas Turbine output is proportional to the air mass flow rate. At higher ambient temperatures, combustion turbines lose capacity, since the mass flow rate of air decreases as the air becomes less dense. This degradation in combustion turbine (CT) output at higher temperatures can be avoided by cooling the inlet air to its compressor.

Khaliq & Kaushik **[3]** analysed that the combined production of different types of energy can lead to a greater efficiency of fuel use compared to separate production. On the other hand it is more difficult to satisfy simultaneously many energy demands. There are more constraints and limitations for combined production compared to separate energy generation.

Kaynakli &Yamankaradeniz **[4]** analyzed the first and second law thermodynamic of a single-stage absorption refrigeration cycle with water/lithium bromide as working fluid pair is performed. Thermodynamic properties of each point in the cycle are calculated using related equation of state. Heat transfer rate of each component in the cycle and some performance parameters are calculated from the first law analysis. From second law analysis, the entropy generation of each component and the total entropy generation of all the system components are obtained.

Kalogirou et al **[5]** studied absorption cooling offers the possibility of using heat to provide cooling. For this purpose heat from a conventional boiler can be used for waste heat and solar energy. When the latter systems are used absorption systems minimize also the adverse effects of burning fossil fuels and thus protect the environment. Absorption systems fall into two major categories, depending on the working fluids. These are the ammonia-water systems, in which ammonia is the refrigerant and lithium bromide-water systems in which water vapor is the refrigerant. The future trends of research in this area would be on other refrigerant pairs which will be more effective.

Hawaj et al **[6]** analyzed cogeneration scheme comprising a combined cycle power plant with an absorption chiller investigating the effect of parameters like turbine inlet temperature, compressor pressure ratio, steam to gas mass flow rate ratio etc.

Talbi &.Agnew **[7]** worked out on exergy analysis of single-effect absorption refrigeration cycle with lithium-bromide-water as the working fluid pair; an optimization procedure that consists of determining the enthalpy, entropy, temperature, mass flow rate, heat rate in each component and coefficient of performance has been performed.

Wang & Chiou **[8]** developed two well-proven techniques, namely steam injection gas turbine (STIG) and inlet air cooling (IAC) are very effective features that can use the generated steam to improve the power generation capacity and efficiency.

Kakaras et al **[9]** analyzed the effect of ambient air temperature variation on the power output and efficiency is given and the results from the integration of an evaporative cooler and of the air-cooling system under consideration are presented and demonstrating the gain in power output and efficiency.

Ali **[10]** has studied simple gas turbine system with inlet air refrigeration by vapour compression cycle. Mass flow rate term has been replaced by volumetric flow rate, thus reducing the inlet air temperature increases the volumetric flow rate .The results shows improvement in terms of cycle efficiency and specific power output.

Khaliq & Kumar **[11]** analyzed the thermodynamic performance of the combustion gas turbine trigeneration system has been studied based on first law as well as second law analysis. The effects of overall pressure ratio and process heat pressure fuel utilization efficiency, electrical to thermal energy ratio, second law efficiency, and exergy destruction in each component are examined. Results for gas turbine cycle, cogeneration cycle, and trigeneration cycle are compared.

Bejan et al **[12]** 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.

Bilgen **[13]** 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.

Khaliq and Kaushik **[14]** 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.

Lucia et al **[15]** 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.

Khaliq **[16]** 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.

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.

Kachhwaha et al **[18]** 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 H2O-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.

Zogou et al **[19]** worked out on transient simulation of a trigeneration system fuelled by natural gas. Efficiency comparison between cogeneration and separate production of electricity and heat carried out by him and a case has been studied on a combined gas turbines – steam turbine process for the cogeneration of electricity and steam in a chemical industry.

Yadav **[20]** worked out the exergy analysis of a novel gas/steam combined cycle employing water/steam cooled gas turbine. Stage by stage component wise exergy loss calculations and analysis is performed. The availability, irreversibility and effectiveness of components have been calculated for both topping and bottoming cycle.

2.2 Conclusions of literature review

The investigation that has been carried out in literature survey is mainly concern with the gas turbine inlet air cooing technique in order to improve its performance under elevated temperature conditions.

The researchers in the field of gas turbine Trigeneration 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. Evaporative cooling can considered whenever the required power increases lies between 8% ant 15%. Furthermore, evaporative cooling to be considered either for peak load turbines or where the turbine is required to work at peak load condition for few hours.

The main conclusions from the Literature review are pointed below:

- Exergy analysis of gas turbine Trigeneration system has been performed by various researchers based on first and second law of thermodynamics in which exergy destruction in each 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 trigeneration system viz. simple cycle, gas turbine trigeneration system with alternative regeneration, gas turbine trigeneration 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 trigeneration plants consisting of a gas turbine, heat recovery steam generator and steam turbine has been worked out.

2.3 Objective of present research work

1. To develop a computer based design methodology for a gas turbine trigeneration system.

2. To explore different techniques that can be used in trigeneration system for enhancing its performance.
3. To perform a parametric study for wide range of variables such as compressor Inlet air temperature, Pressure ratio, Turbine Inlet temperature etc and the effect of these parameters on First law efficiency, Second law efficiency, Fuel utilization efficiency, Specific fuel consumption and Process steam etc. Quantity and exergy destruction in a component has been studied.

Chapter 3

FORMULATION FOR TRIGENERATION SYSTEM

In this chapter 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) for analysis, 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 as shown in Figure 3.1. A set of governing equations for a particular component (k) is expressed as:

Mass rate balance

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

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 exergy can be expressed as

$$E_k = E_k^{PH} + E_k^{CH} \tag{3.4}$$

where the chemical exergy of air, fuel and water can be found in Appendix. The chemical exergy of gases mixture is obtained by summing the overall compositions of air that includes N_2, O_2, CO2, H_2O(g) and other gases.

3.1 System description

A schematic diagram of Trigeneration system with various significant components is shown in Figure 3.1. It is based on Brayton cycle and vapor absorption refrigeration cycle comprising of an axial flow compressor, combustion chamber, turbine, counter current heat exchanger called heat recovery steam generator and vapor absorption refrigeration system.

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 & fuel named gases enters the turbine where the gases are expanded and produce the work output. The heat carried by the exhaust gases is recovered in the HRSG to generate the steam. Now the exhaust gases from HRSG enter the generator of vapour absorption refrigeration system where it gives the heat quantity to obtain effect.

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 set to be 130˚C to prevent the possibility of vapour condensation. Because formation of vapour condensation leads to formation of sulphuric acid that causes the acidic corrosion.

A typical single effect 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.

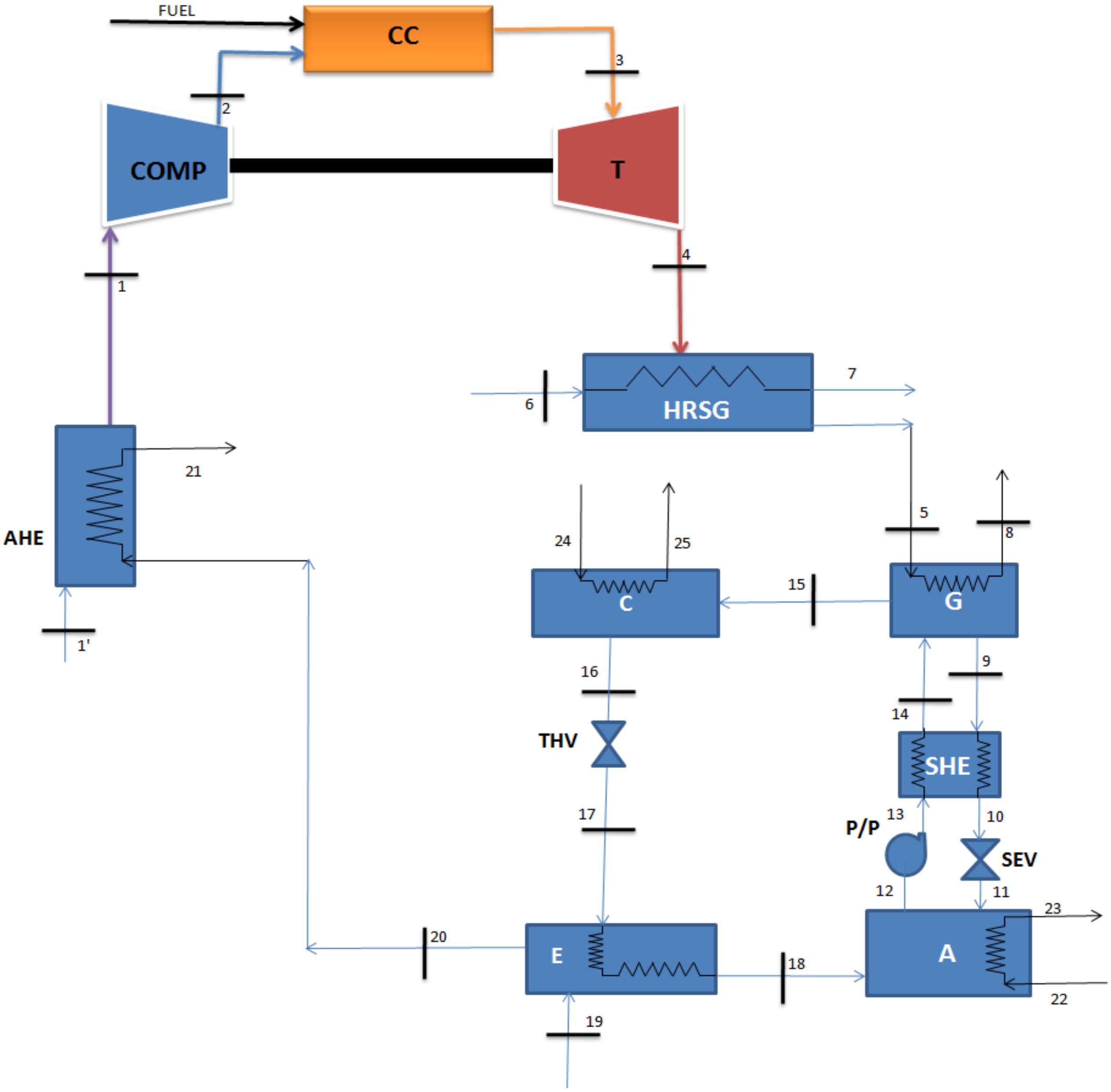

Fig 3.1 Schematic of trigeneration system

State points in Figure 3.1-

1	Inlet air to Compressor
2	Compressor outlet or combustion chamber inlet
3	Combustion chamber outlet or turbine inlet
4	Exit from Gas Turbine or Entry to HRSG
5	Exit from HRSG or Inlet to Generator of VARS
6	Feed water inlet to HRSG
7	Steam outlet from HRSG
8	Inlet to solution heat exchanger
9	Exit from solution heat exchanger
10	Inlet to Pressure Reduction Valve
11	Exit from Pressure Reduction Valve or Inlet to absorber
12	Exit from Absorber or inlet to solution pump
13	Exit from solution Pump
14	Exit from solution heat exchanger
15	Inlet to condenser
16	Exit from condenser
17	Exit from throttle valve
18	Exit of evaporator
19	Cooling water inlet to evaporator
20	Chilled water outlet from evaporator
21	Exit of air heat exchanger
22	Cooling water inlet to absorber
23	Cooling water outlet from absorber
24	Cooling water inlet to condenser
25	Cooling water outlet from condenser

f fuel input to combustion chamber

1’ Inlet air to air heat exchanger

3.2 Assumptions made for mathematical formulation

The mathematical formulation of the present analysis is based on the following assumptions:

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. 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.
7. The pressure drop in vapour absorption refrigeration system is neglected.
8. The HRSG unit is a single pressure counter current heat exchanger with fixed effectiveness.

3.3 Analysis of system components

3.3.1 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.

The air is compressed by the air compressor and transferred to the combustion chamber in order to mix with fuel. The energy is absorbed by the compressor in the form of work is given by

$$\dot{W}_C = \dot{n}_a(\overline{h}_1 - \overline{h}_2) = \dot{m}_a \frac{\overline{h}_1 - \overline{h}_2}{M_a} \tag{3.5}$$

Where n_a is molar flow rate of air, m_a is mass flow rate of air, M_a is molecular weight of air and h_2-h_1 is the difference of enthalpy between states 1 and 2.

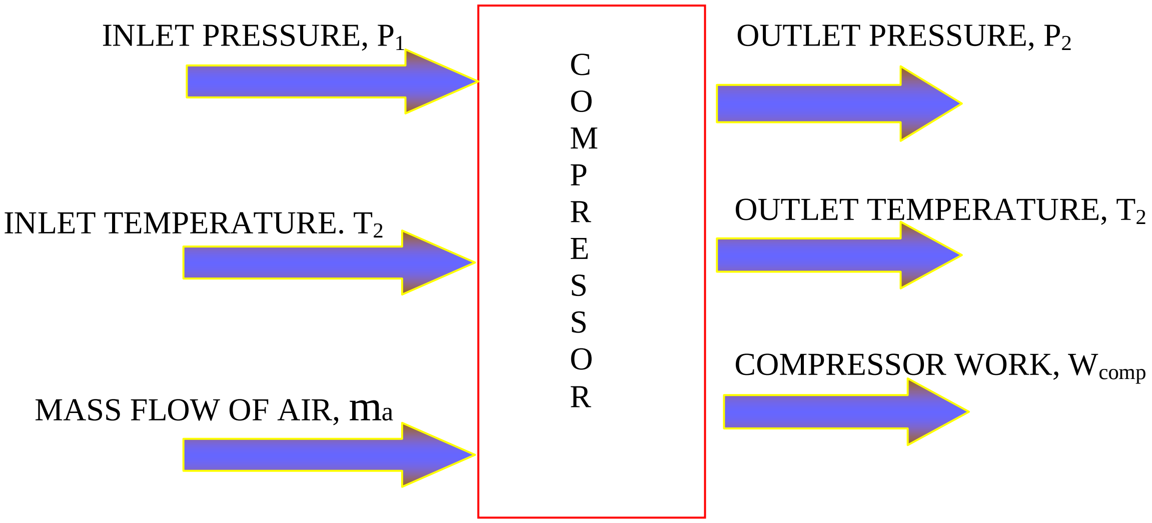

Fig: 3.2 Compressor information flow diagram

An information flow diagram of compressor is shown in Figure 3.2, across a compressor stage the temperature rise is:

$$\frac{T_2}{T_1} = \frac{\left(\frac{P_2}{P_1}\right)^{\frac{(\gamma-1)}{\gamma}}}{\gamma \times \eta} \tag{3.6}$$

Where γ is adiabatic index, η is isoentropic efficiency of compressor.

The above equation holds true if the physical properties of working fluids is constant. But in real they vary with temperature & pressure.

To account for this we assume that the total pressure rise is occurring through a large number of stages, i.e. 17 for the axial flow compressor. The pressure and temperature rise across a stage is very small so the properties of working fluid assumed constant.

Each step properties of working fluid is calculated corresponding to pressure and temperature. In this way, the above equations evaluated across all stages and the summation of work across all stages gives the total compressor work. Final stage output pressure and temperature are used as the input to combustor.

$$r_C = (T_2/T_1)^{(\gamma/\gamma-1)} \tag{3.7}$$

$$\eta_C = (\bar{h}_{2s} - \bar{h}_1)/(\bar{h}_2 - \bar{h}_1) \tag{3.8}$$

r_C is the pressure ratio of compressor, η_C is the isentropic efficiency of compressor and $\bar{h}_{2s}$is the isentropic enthalpy at outlet of compressor. Since ϒ is a function of temperature, T_{2s} is determined in the following way instead of equation (3.7) .

Change in entropy during isentropic compression in compressor

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

$$+ X_{1,O2}\left[\overline{s}^0(T_{2s}) - \overline{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{O_2}$$

$$+ X_{1,CO2}\left[\overline{s}^0(T_{2s}) - \overline{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{CO_2}$$

$$+ X_{1,H2O}\left[\overline{s}^0(T_{2s}) - \overline{s}^0(T_1) - \bar{R}\ln\frac{p_2}{p_1}\right]_{H_2O}$$

$$=0 \tag{3.9}$$

where $X_{1,N2}$ =mole fraction of N_2 in air, $X_{1,O2}$ =mole fraction of O_2 in air,

$X_{1,CO2}$ =mole fraction of CO_2 in air, $X_{1,H2O}$ =mole fraction of H_2O in air.

Using the inbuilt specific entropy expressions for N_2, O_2, CO_2 and H_2O from software, the above expression is solved for T_{2s} and corresponding enthalpy of h_{2s} is determined.

Physical Exergy at state 1-

At this state $T_1 = T_0$ and $p_1=p_0$. Accordingly, $h_0 = h_1$ and $s_1 = s_0$ and the physical exergy component vanishes:

$$\dot{E}_1^{PH} = 0 \tag{3.10}$$

Chemical Exergy at state 1-

$$\dot{E}_1^{CH} = 0 \tag{3.11}$$

Total Exergy at state 1-

$$\dot{E}_1 = \dot{E}_1^{PH} + \dot{E}_1^{CH} = 0 \tag{3.12}$$

Physical Exergy at state 2-

$$\dot{E}_2^{PH} = \dot{m}_{a2} \frac{\bar{h}_2 - \bar{h}_0 - T_0(\bar{s}_2 - \bar{s}_0)}{M_a} \tag{3.13}$$

Chemical Exergy at state 2-

$$\dot{E}_2^{CH} = 0 \tag{3.14}$$

Total Exergy at state 2-

$$\dot{E}_2 = \dot{E}_2^{PH} + \dot{E}_2^{CH} \tag{3.15}$$

Exergy Destruction in Compressor-

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

$$\dot{E}_{D,C} = \dot{E}_1 - \dot{E}_2 - \ddot{W}_C \tag{3.16}$$

3.3.2 Combustion Chamber

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

$$\frac{\dot{n}_f}{\dot{n}_a} = \lambda \tag{3.17}$$

and $$\frac{\dot{n}_p}{\dot{n}_a} = 1 + \lambda \tag{3.18}$$

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

For complete combustion of natural gas (methane), chemical equation takes the following form

$$\lambda CH_4 + \left[X_{1,N2}N_2 + X_{1,O2}O_2 + X_{1,CO2}CO_2 + X_{1,H2O}H_2O\right]$$

$$\rightarrow [1+\lambda] \quad \left[X_{2,N2}N_2 + X_{2,O2}O_2 + X_{2,CO2}CO_2 + X_{2,H2O}H_2O\right] \tag{3.19}$$

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

Mole fraction of N_2 $$X_{2,N2} = \frac{X_{1,N2}}{1+\lambda} \tag{3.20}$$

Mole fraction of O_2 $$X_{2,O2} = \frac{X_{1,O2} - 2\lambda}{1+\lambda} \tag{3.21}$$

Mole fraction of CO_2 $$X_{2,CO2} = \frac{X_{1,CO2} + \lambda}{1+\lambda} \tag{3.22}$$

Mole fraction of H_2O $$X_{2,H2O} = \frac{X_{1,H2O} + 2\lambda}{1+\lambda} \tag{3.23}$$

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.

$$0 = \dot{Q}_{cv} - \cancel{\dot{W}_C} + \dot{n}_f \overline{h_f} + \dot{n}_a \overline{h_a} - \dot{n}_P \overline{h_P}$$

$$0 = \dot{Q}_{cv} + \dot{n}_f \overline{h_f} + \dot{n}_a \overline{h_a} - \dot{n}_P \overline{h_P} \tag{3.24}$$

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

$$\dot{Q}_{cv} = -l_f n_f \eta_{cc} \overline{LHV} \tag{3.25}$$

$$= \dot{n}_a \left(- l_f \lambda \eta_{cc} \overline{LHV}\right) \tag{3.26}$$

Where l_f =0.02, LHV=Lower Heating Value, η_{cc} =Efficiency of combustion chamber

Combining equation (3.23) to (3.26)

$$0 = -l_f \lambda \eta_{cc} \overline{LHV} + \bar{h}_a + \lambda \bar{h}_f - (1+\lambda)\bar{h}_p \tag{3.27}$$

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

$$\bar{h}_a = \left[X_{1,N2}N_2 + X_{1,O2}O_2 + X_{1,CO2}CO_2 + X_{1,H2O}H_2O\right]_{T2} \tag{3.28}$$

$$(1+\lambda)\bar{h}_p = \begin{bmatrix} X_{1,N2}h_{N_2} + (X_{1,O2} - 2\lambda)\bar{h}_{O_2} + (X_{1,CO2} + \lambda)\bar{h}_{CO_2} \\ + (X_{1,H2O} + 2\lambda)\bar{h}_{H_2O} \end{bmatrix}_{T_3} \tag{3.29}$$

Combining the last three equations and solving for λ yields

$$\lambda = \frac{[X_{1,N2}\Delta\bar{h}_{N_2} + X_{1,O2}\Delta\bar{h}_{O_2} + X_{1,CO2}\Delta\bar{h}_{CO_2} + X_{1,H2O}\Delta\bar{h}_{H_2O}]}{[\bar{h}_p - 0.02\eta_{CC}\overline{LHV} + \left(2\bar{h}_{O_2} + \bar{h}_{CO_2} + 2\bar{h}_{H_2O}\right)]_{T3}} \tag{3.30}$$

Mass flow rate of air is given by

$$\dot{m}_a = \frac{M_a\dot{W}_C}{(1+\lambda)\left(\bar{h}_3 - \bar{h}_4\right) + \left(\bar{h}_1 - \bar{h}_2\right)} \tag{3.31}$$

and Mass flow rate of fuel:

$$\dot{m}_f = \lambda \frac{M_f}{M_a}\dot{m}_a \tag{3.32}$$

where M_f = Molecular weight of fuel (Methane)

M_a = Molecular weight of air

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

Physical Exergy of fuel (methane) at state f-

$$\dot{E}_f^{PH} = \dot{m}_f\left[\bar{h}_f - \bar{h}_0 - T_0\left(\bar{s}_f - \bar{s}_0\right)\right] \tag{3.33}$$

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

$$\dot{E}_f^{PH} = \dot{m}_f RT_1 \ln\frac{p_f}{p_1} \tag{3.34}$$

Chemical Exergy of fuel (methane) at state f-

$$\dot{E}_f^{CH} = \dot{m}_f \times \frac{e_{methane}^{CH4}}{M_f} \times 10^{-3} \tag{3.35}$$

where $e_{methane}^{CH4}$=424348 kJ/kmol

Total Exergy at state f-

$$\dot{E}_f = \dot{E}_f^{PH} + \dot{E}_f^{CH} \tag{3.36}$$

Physical Exergy at state 3-

$$\dot{E}_3^{PH} = \dot{m}_3 \frac{\overline{h}_3 - \overline{h}_0 - T_0(\overline{s}_3 - \overline{s}_0)}{M_p} \tag{3.37}$$

At state 3, there are combustion products (flue gases) after combustion. The restricted dead state corresponding to the mixture at state 3 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 is determined for state 3. The contribution of the gas phase to the chemical exergy is evaluated as

$$\sum(x'_k \times e_k^{CH}) + R\ To \sum(x'_k \times ln x'_k) \tag{3.38}$$

On the basis of 1 kmol of mixture at state 3, we have (x_{1N2} + x_{1O2} +x_{1CO2}+x_{1H2O}) kmol as a gas phase and 1-(x_{1N2}+x_{1O2}+x_{1CO2}+x_{1H2O}) kmol as liquid water where x_{1N2}, x_{1O2}, x_{1CO2} and x_{1H2O} is initial mole fraction of air, thus

Chemical exergy at state 3-

$$\dot{E}_4^{CH} = (\dot{m}_3 / M_p) \times [(x_{1N2} + x_{1O2} + x_{1CO2} + x_{1H2O}) \times \sum(x'_k \times e_k^{CH}) +$$

$$\bar{R}\ To \sum(x'_k\ ln x'_k) \times (1 - (x_{1N2} + x_{1O2} + x_{1CO2} + x_{1H2O})) \times e_{ow}] \times 10^{-3} \tag{3.39}$$

Exergy Destruction in Combustion Chamber-

$$\dot{E}_{D,CC} = \sum_i \left(1 - \cancel{\frac{T_0}{T_1}}\right) \dot{Q}_1 - \cancel{\dot{W}_C} + \dot{E}_2 + \dot{E}_f - \dot{E}_3$$

$$\dot{E}_{D,CC} = \dot{E}_2 + \dot{E}_f - \dot{E}_3 \quad (3.40)$$

3.3.3 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

$$\left[X_{2,N2}N_2 + X_{2,O2}O_2 + X_{2,CO2}CO_2 + X_{2,H2O}H_2O\right]_{T4} \quad (3.41)$$

The value of specific enthalpy h_4 can be evaluated using the turbine isentropic efficiency. Solving the expression for expansion for turbine isentropic efficiency

$$\eta_{st} = \frac{\overline{h}_3 - \overline{h}_4}{\overline{h}_3 - \overline{h}_{4s}} \quad (3.42)$$

We get

$$\overline{h}_4 = \overline{h}_3 - \eta_{st}\left(\overline{h}_3 - \overline{h}_{4S}\right) \quad (3.43)$$

where η_{st} is isentropic efficiency of turbine, h3 and h_4 are enthalpies of flue gas at state 3 and 4. H_{4s} is the isentropic enthalpy at turbine exit.

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

$$\begin{aligned}\overline{s^0}_{4S} - \overline{s}_3 = {} & X_{2,N2}\left[\overline{s}^0(T_{4s}) - \overline{s}^0(T_3) - \overline{R}\ln\frac{p_4}{p_3}\right]_{N_2} \\ & + X_{2,O2}\left[\overline{s}^0(T_{4s}) - \overline{s}^0(T_4) - \overline{R}\ln\frac{p_4}{p_3}\right]_{O_2} \\ & + X_{2,CO_2}\left[\overline{s}^0(T_{4s}) - \overline{s}^0(T_4) - \overline{R}\ln\frac{p_4}{p_3}\right]_{CO_2} \\ & + X_{2,H2O}\left[\overline{s}^0(T_{4s}) - \overline{s}^0(T_4) - \overline{R}\ln\frac{p_4}{p_3}\right]_{H_2O} \\ & = 0 \end{aligned} \quad (3.44)$$

Physical Exergy at state 4-

$$\dot{E}_4^{PH} = \dot{m}_3 \frac{\overline{h}_3 - \overline{h}_0 - T_0(\overline{s}_4 - \overline{s}_0)}{M_P} \tag{3.45}$$

Exergy Destruction in Turbine-

$$\dot{E}_{D,turbine} = \sum_i \left(1 - \cancel{\frac{T_0}{T_1}}\right) \dot{Q}_1 + \dot{W}_T + \dot{E}_3 - \dot{E}_4 \tag{3.46}$$

3.3.4 Heat recovery steam generator

The waste heat recovery process in heat recovery steam generator is illustrated on temperature profile diagram in fig 4.2 In this process water enters the boiler in the form of compressed liquid at feed water temperature T_{fw}. 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 turbine 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.

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}).

$$T_P = T_{Drum} + PP$$

$$T_{ECO} = T_{Drum} - AP$$

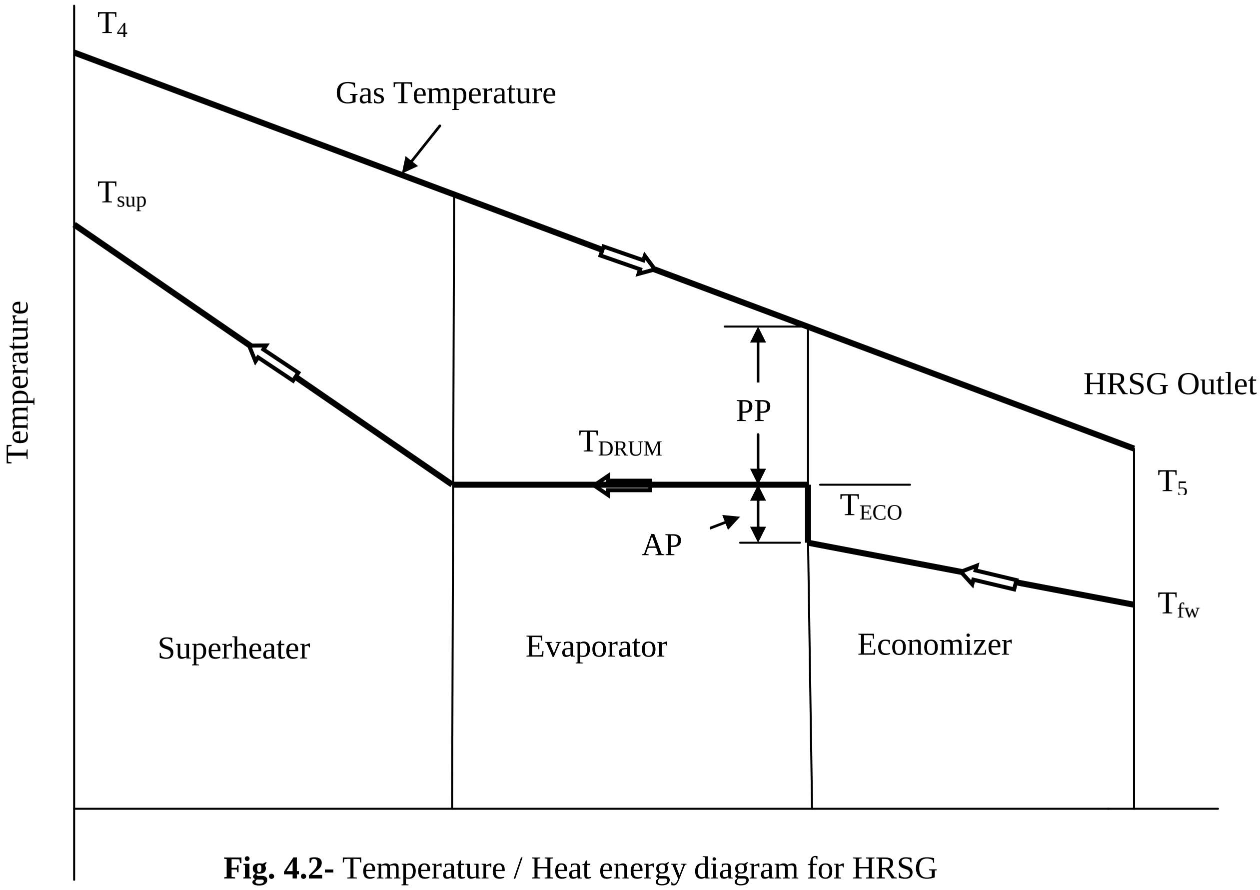

Fig. 4.2- Temperature / Heat energy diagram for HRSG

The steam generated (m_{st}) for each kg/s of exhaust gases can be determined by applying the mass and energy conservation principles across superheater and evaporator as-

$$\dot{m}_{st} \times (\bar{h}_{sup} - \bar{h}_{fw}) = \dot{m}_p \times (\bar{h}_4 - \bar{h}_5) \qquad (3.47)$$

where $\bar{h}_{sup}$ is the enthalpy of superheated steam generated in HRSG, $\bar{h}_{fw}$ is enthalpy of feed water at inlet to HRSG, m_p is mass flow rate of gases, $\bar{h}_4$ and $\bar{h}_P$ is enthalpy of flue gases at inlet and pinch point of HRSG respectively

The above equation can be solved for h_7 and T_7 can be calculated iteratively.

Physical Exergy at state 5-

$$\dot{E}_5^{PH} = \dot{m}_5\left[\bar{h}_5 - \bar{h}_0 - T_0\left(\bar{S}_5 - \bar{S}_0\right)\right] \tag{3.48}$$

Chemical Exergy at state 5-

$$\dot{E}_5 CH = \dot{m}_w \times \frac{e_{water}}{M_w} \times 10^{-3} \tag{3.49}$$

where e_{water}=45 kJ/kmol

Physical Exergy at state 6-

$$\dot{E}_6^{PH} = \dot{m}_6\left[\bar{h}_6 - \bar{h}_0 - T_0\left(\bar{S}_6 - \bar{S}_0\right)\right] \tag{3.50}$$

Exergy Destruction in HRSG-

$$\dot{E}_{D,HRSG} = \dot{E}_6 - \dot{E}_7 + \dot{E}_5 - \dot{E}_6 \tag{3.51}$$

The component of VARS is discussed below

3.3.5 Generator

The exhaust gases from HRSG enters the generator of Vapour Absorption Refrigeration System where exhaust gases gives heat to the solution in generator the heat transfer equation are as follows

$$\dot{Q}_g = \dot{n}_p(\bar{h}_5 - \bar{h}_8) \tag{3.52}$$

$$f_c = [\{1 - \left(1 - \frac{X_9}{100}\right)\}/\{\left(1 - \frac{X_{12}}{100}\right) - \left(1 - \frac{X_9}{100}\right)\}] \tag{3.53}$$

Here, Qg is the heat given by the exhaust gases to the generator, f_c is the circulation rate.

The exergy destruction in the generator is given by following equation

$$\dot{E}_{D,g} = \dot{E}_5 + \dot{E}_{14} - \dot{E}_9 - \dot{E}_8 - \dot{E}_{15} \tag{3.54}$$

3.3.6 Absorber

From absorber the strong solution enters the generator through pump &solution heat exchanger and weak solution from generator enters to absorber also the refrigerant in vapor state enters to absorber so mixing process takes place in absorber.

The heat transfer equation for absorber given below

$$\dot{Q}_{abs} = \dot{m}_r\,(\bar{h}_{18} - \bar{h}_{11}) + f_C\,(\bar{h}_{11} - \bar{h}_{12}) \tag{3.55}$$

where f is the circulation rate & m_r is the mass of refrigerant

The exergy destruction in absorber is given by the following equation

$$\dot{E}_{D,Abs} = \dot{E}_{11} + \dot{E}_{18} + \dot{E}_{22} - \dot{E}_{12} - \dot{E}_{23} \tag{3.56}$$

3.3.7 Solution Heat Exchanger

In solution heat exchanger the weak solution in hot state from generator enters it and heat up the strong solution which enters in cold state the heat exchange between two solutions is given by the following equation

$$\dot{Q}_{SHX} = \dot{m}_1\,(\bar{h}_{14} - \bar{h}_{13}) \tag{3.57}$$

where m_1 is mass flow rate of strong solution

The exergy destruction in solution heat exchanger is given by the following equation

$$\dot{E}_{D,SHX} == \dot{E}_9 + \dot{E}_{13} - \dot{E}_{14} - \dot{E}_{10} \tag{3.58}$$

3.3.8 Solution Pump

The function of the solution pump is that it transfers the solution from absorber (low pressure region) to generator (high pressure region).The work consumed by the pump is given by following equation

$$\dot{W}_{Pump} = V \times \left(P_g - P_a\right) \times \dot{m}_1/\eta_{pump} \tag{3.59}$$

$$= \dot{m}_1 \times (\bar{h}_{12} - \bar{h}_{11}) \tag{3.60}$$

Where, $\dot{m}_1 = f_C \times \dot{m}_r$

η_{pump} is the efficiency of pump, P_g &P_a be the pressure of generator and absorber, m_1 is the mass of strong solution enters the pump & V be the volume of strong solution entering the pump.

3.3.9 Condenser

The refrigerant in steam state enters to the condenser where it condenses and exchanging heat with water which passes through the condenser the heat transfer equation for condenser given below

$$\dot{Q}_{cond} = \dot{m}_r \times (\bar{h}_{15} - \bar{h}_{16}) \tag{3.61}$$

The exergy destruction in condenser is given by the following equation

$$\dot{E}_{D,Cond} = \dot{E}_{15} - \dot{E}_{16} + \dot{E}_{24} - \dot{E}_{25} \tag{3.62}$$

3.3.10 Evaporator

The condensed refrigerant from condenser after throttling enters to the evaporator where it evaporates and also the air from ambient is passed through the evaporator where it cools and then it is send to compressor of gas turbine unit, the heat transfer equation for evaporator is given below

$$\dot{Q}_{evap} = \dot{m}_r \times (\bar{h}_{17} - \bar{h}_{16}) \tag{3.63}$$

The exergy destruction in evaporator is given by the following equation

$$\dot{E}_{D,Evap} = \dot{E}_{19} + \dot{E}_{17} - \dot{E}_{18} - \dot{E}_{20} \tag{3.64}$$

3.4 Performance Parameters

The relevant parameters required for the combined first and second law analysis of gas turbine cogeneration system are summarised below:

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. This reflects the first law of thermodynamics, which is concern with quantity not energy quality. It is also known as energetic or fuel utilization efficiency.

$$\eta_I = (\dot{W}_{el} + \dot{Q}_{process} + \dot{Q}_{evap})/\dot{Q}_{fuel} \quad (3.65)$$

Where $\dot{W}_{el}$=Electrical work rate, $\dot{Q}_{process}$=Process heat rate, $\dot{Q}_{fuel}$= evaporator Cooling rate

Second –Law Efficiency (η_{II}): Since electrical power is more valuable than process heat according to the second law of thermodynamics unlike energy, the exergy is always destroyed in any real process. 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 defined as:

$$\eta_{II} = (\dot{W}_{el} + \dot{E}_{process} + \dot{E}_{evap})/\dot{E}_{fuel} \quad (3.66)$$

Where $\dot{E}_{process}$ = process exergy rate, $\dot{E}_{evap}$ = exergy rate in evaporator

$\dot{E}_{fuel}$ =fuel exergy rate

Electrical to Thermal Energy Ratio (R_{ET}): The cost effectiveness of any trigeneration system is directly related to the amount of power it can produce for a given amount of process heat and cold needed. Thus the electrical to thermal energy ratio (RET) is an important parameter used to assess the performance of such a system. Which is defined as

$$R_{ET} = \dot{W}_{el}/(\dot{Q}_{process} + \dot{Q}_{evap}) \quad (3.67)$$

Exergy Destruction Ratio*:* 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.

$$y_D = \dot{E}_D / \dot{E}_{f,tot} \quad (3.68)$$

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

$$y_D^* = \dot{E}_D / \dot{E}_{f,tot} \tag{3.69}$$

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.

Exergy Loss Ratio: It is 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

$$y_L = \dot{E}_L / \dot{E}_{f,tot} \tag{3.70}$$

Exergetic Efficiency: It is a parameter for evaluating thermodynamic performance. The exergetic efficiency (second-law efficiency, effectiveness, or rational efficiency) provides a true measure of the performance of an energy system from the thermodynamic viewpoint.

As an illustration, let us consider a system at steady state where, in terms of exergy, the rates at which the fuel is supplied and the product is generated are E_F and E_P, respectively. An exergy rate balance for the system reads

$$\dot{E}_F = \dot{E}_P + \dot{E}_D + \dot{E}_L \tag{3.71}$$

Where $\dot{E}_D$ and $\dot{E}_L$ denotes the rate of exergy destruction and exergy loss respectively

The exergetic efficiency η_{exgt} is the ratio between product and fuel,

$$\eta_{exgt} = \frac{\dot{E}_p}{\dot{E}_f}$$

$$= 1 - \frac{\dot{E}_D + \dot{E}_L}{\dot{E}_f} \tag{3.72}$$

The exergetic efficiency shows the percentage of the fuel exergy provided to a system that is found in the product exergy.

Chapter 4

RESULTS AND DISCUSSION

Chapter describes the result and discussion of case study for cogeneration power system (full load and part load), trigeneration system and parametric studies followed by model validation

4.1 Model validation for cogeneration system with regeneration

The computer program developed for cogeneration system has been validated with the example given in [1]. The various component of gas turbine cogeneration system is shown in figure 4.1

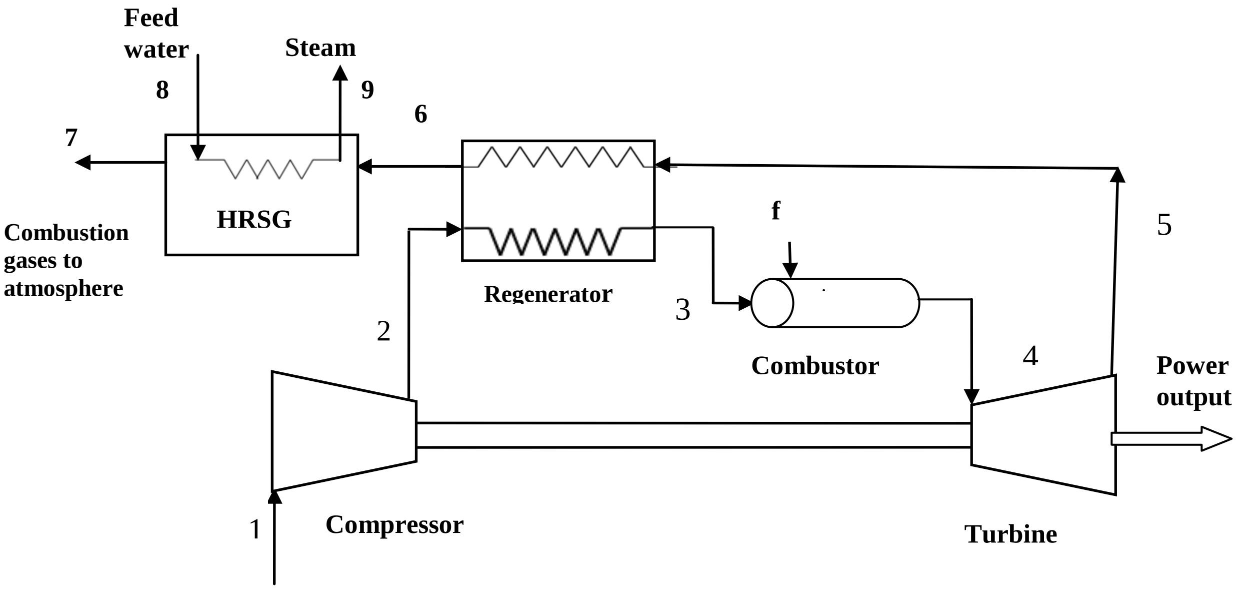

Figure 4.1: Gas turbine cogeneration system with regeneration

Systems Description

A standard cogeneration System with regeneration is considered for the present analysis. The diagram is shown in figure. 3.1 The system works on Brayton cycle with CH_4 – O_2 combustion and a HRSG utilizing the waste heat of the cycle. The cycle is comprised of an air compressor, a regenerator, a combustor and a turbine. Air after compression in the compressor enters the regenerator where its temperature is raised by the exhaust gases from turbine. After regenerator it enters into combustion chamber where its temperature is raised by the combustion of fuel.

The gases then expand in the turbine and produce the work output (alternator to generate electricity). The heat carried by the exhaust gases is recovered in the regenerator and HRSG to increase the temperature of air and generate steam respectively.

The various input parameters [1] supplied to the program are given in Table 4.1

Table 4.1: Input data for analysis of cogeneration system

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
Isentropic efficiency of compressor (η_{SC}), in %	0.85
Turbine inlet temperature (TIT) or maximum cycle temperature (T_4), in K	1500
Isentropic efficiency of Turbine (η_{GT}), in %	0.86
Exhaust pressure of combustion prod after HRSG (P_7), in bar	1.113
Mass flow rate of steam generated in HRSG (m_{st}) in kg/s	14
Pressure of steam generation (P_5) in bar	20
Pressure of feed water at inlet of HRSG (P_6), in bar	24
Temperature of feed water at inlet to HRSG (T_6), in K	298.15
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	91.34 (91.27)	298.15	1.013
2	Air	91.34 (91.27)	615.1 (603.73)	10.13 (10.13)
3	Air	91.34 (91.27)	850	9.62 (9.62)
4	Combustion Products	92.99 (92.91)	1500	9.14 (9.14)
5	Combustion products	92.99 (92.91)	1009.9 (1006.16)	1.09 (1.09)
6	Combustion products	92.99 (92.91)	796.2 (779.88)	1.06 (1.06)
7	Combustion products	92.99 (92.91)	435.3 (426.89)	1.013
8	Water	14	298.15	20
9	Steam	14	487.1 (485.57)	20
f	Methane	1.652 (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 (100.083)	0.6046 (0.6069)	100.811 (100.689)
5	Combustion products	-9264 (-8839)	238.1	37.650 (37.4158)	0.6046 (0.6069)	38.255 (38.0227)
6	Combustion products	-16586 (-16522.68)	230.2	21.408 (21.3851)	0.6046 (0.6069)	22.012 (21.992)
7	Combustion products	-28123 (-27974.5)	211.2	2.172 (2.1061)	0.6046 (0.6069)	2.776 (2.713)
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)
10	Methane	-74872	-	0.6267 (0.6267)	84.35 (84.366)	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	25.372 (25.48)	64.36 (64.56)	29.85 (29.98)
HRSG	6.456 (6.23)	16.37 (15.78)	7.59 (7.33)
Gas turbine	2.959 (3.01)	7.50 (7.63)	3.48 (3.54)
Air preheater	2.542 (2.63)	6.44 (6.66)	2.99 (3.09)
Air compressor	2.093 (2.12)	5.30 (5.37)	2.46 (2.49)
Overall plant	39.422 (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 is clear that the output data of present model matches reasonably well with the values of reference [1] given in the bracket.

4.2 Case study for gas turbine power plant

The formulation of gas turbine Trigeneration system has been discussed in chapter 3. In order to analyze the system two cases have been considered.

i. Cogeneration power plant
ii. Trigeneration power plant

4.2.1 Case –I (Cogeneration power plant at full load)

System description

A schematic diagram of Cogeneration plant with various significant components is shown in figure.4.2. It is based on Brayton cycle comprising of an axial flow compressor, combustion chamber turbine and counter current heat exchanger called heat recovery steam generator. The air at atmospheric condition enters to the air compressor, from where the air is compressed to high pressure 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 & fuel named gases enters the turbine where the gases are expanded and produce the work output. The heat carried by the exhaust gases is recovered in the HRSG to generate the steam.

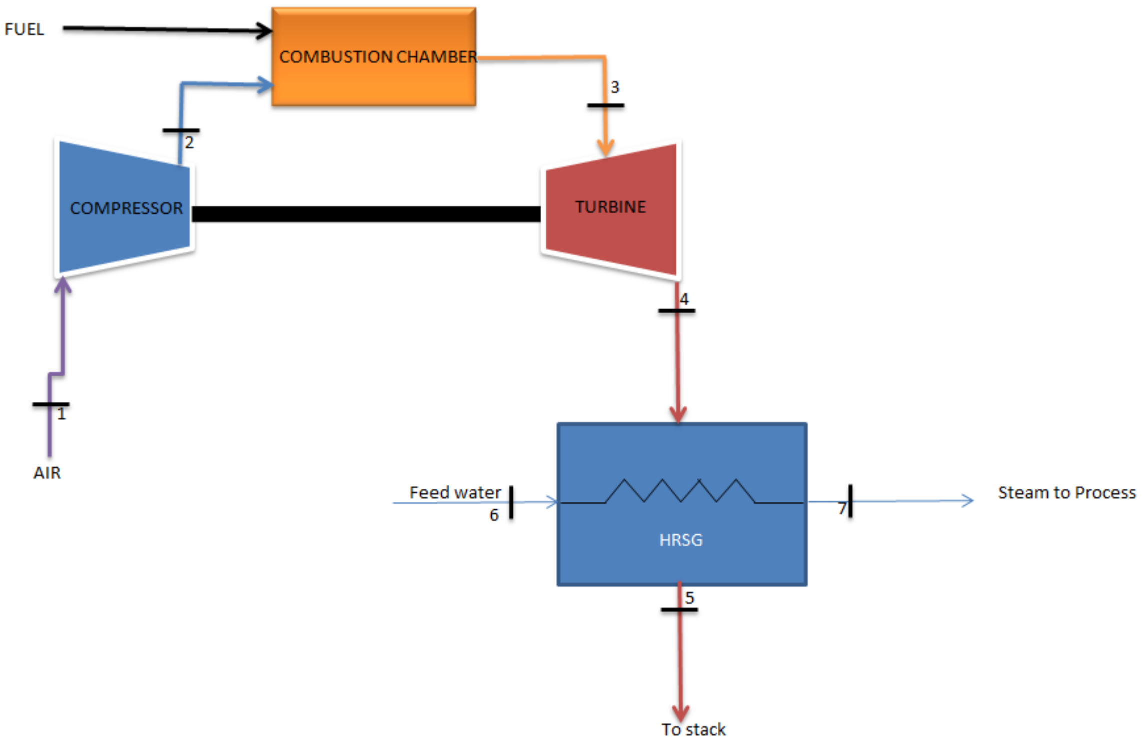

Fig 4.2 Schematic of cogeneration system

Input data for analysis of the cogeneration system shown in Fig 4.2 is given in Table 4.5

Table 4.5 Input data for analysis of cogeneration plant(full load)

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
Isentropic efficiency of compressor (η_{SC}), in %	0.85
Turbine inlet temperature (TIT) or maximum cycle temperature (T_3), in K	1527
Isentropic efficiency of Turbine (η_{GT}), in %	0.86
Exhaust pressure of combustion prod after HRSG (P_5), in bar	1.14
Mass flow rate of steam generated in HRSG (m_{st}) in kg/s	27.77
Pressure of steam generation (P_7) in bar	90
Pressure of feed water at inlet of HRSG (P_6), in bar	90
Temperature of feed water at inlet to HRSG (T_6), in K	383
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 HRSG on the gas side, in %	5

Table 4.6 Operating parameters at state points of the cogeneration system

State	Substance	Mass flow rate (kg/s)	Temperature (K)	Pressure (bar)
1	Air	85.65	298.15	1.013
2	Air	85.65	611.34	10
3	Combustion Products	87.71	1520	9.61
4	Combustion products	87.71	1005.5	1.14
5	Combustion products	87.71	507.9	1.083
6	Water	27.77	383	90
7	Steam	27.77	760.1	90
f	Methane	2.062	311	16

Table 4.7 Calculated parameters at state points of the cogeneration system

State	Substance	Enthalpy (kJ/kmol)	Entropy (kJ/kmolK)	Physical exergy (MW)	Chemical exergy (MW)	Total exergy (MW)
1	Air	-4713	199.9	0	0	0
2	Air	4632	201.3	27.619	0	27.619
3	Combustion Products	694.5	235.2	102.905	1.073	103.978
4	Combustion products	-17309	238.4	40.226	1.073	41.299
5	Combustion products	-32258	217	10.5048	1.073	11.578
6	Water	8428	210.5	0.616	0.041	0.647
7	Steam	60598	102.3	8.7548	0.041	8.791
f	Methane	-74875	-	0.837	112.691	113.528

Table 4.8 Exergy destruction data of gas turbine cogeneration system

Component	Exergy Destruction Rate(MW)	Exergy Destruction (Percentage a)	Exergy Destruction (Percentage b)
Combustion chamber	36.970	77.31	32.56
Air compressor	2.020	4.22	1.77
Gas turbine	2.847	5.95	2.50
HRSG	5.98	12.5	5.26
Overall plant	47.817	100	42.09

As per above analysis it can be concluded that maximum exergy destruction occurred in combustion chamber is shown in Figure 4.3 on the other hand minimum exergy destruction occurred in air compressor. The maximum exergy destruction of percentage exergy destruction with respect to fuel exergy as well as percentage exergy destruction with respect to overall plant occurred in combustion chamber that is shown in figure 4.4 and figure 4.5 respectively.

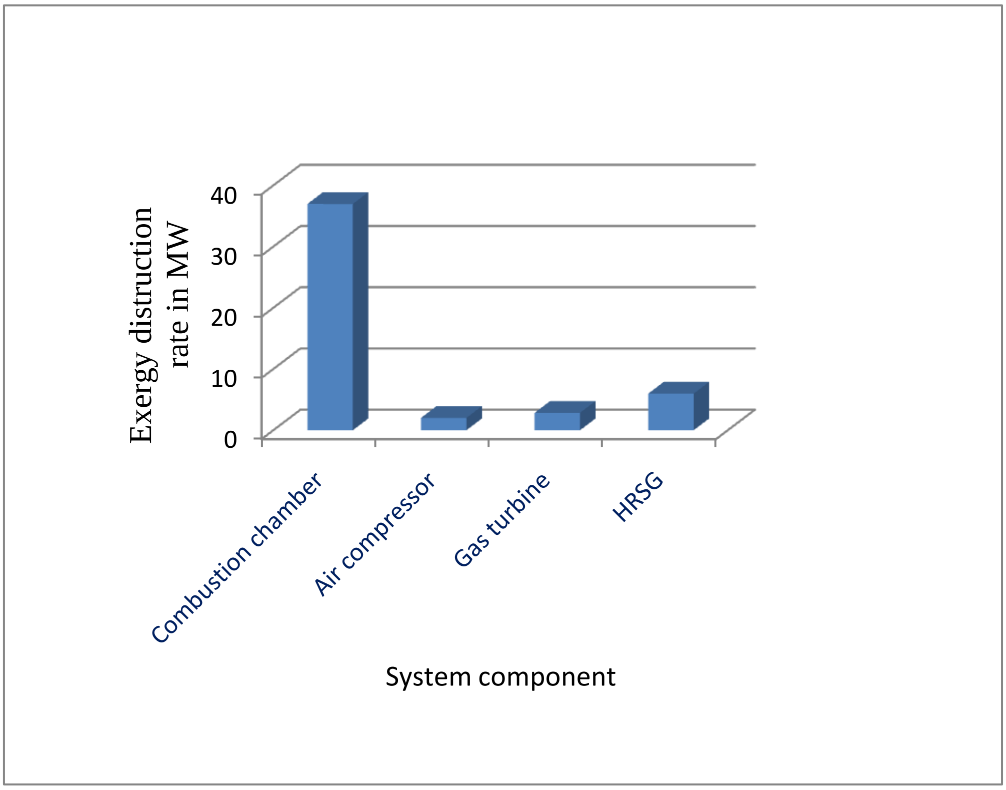

Fig. 4.3 Variation of exergy destruction for cogeneration system component

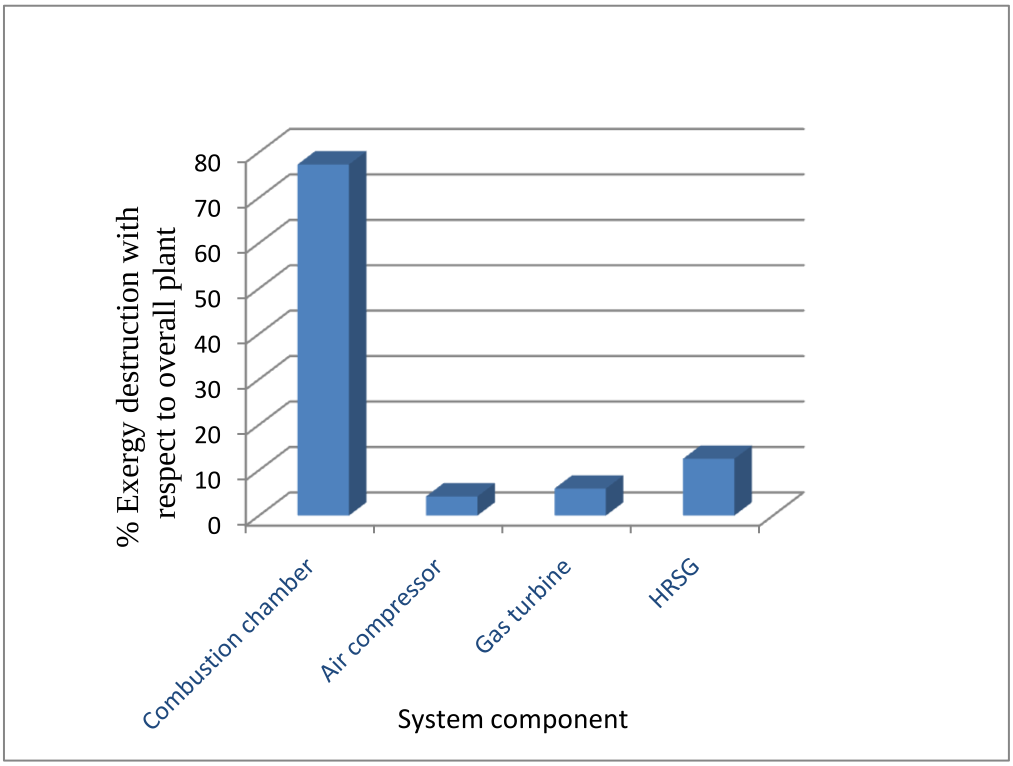

Fig. 4.4 Variation of percentage exergy destruction with respect to overall plant for cogeneration system component

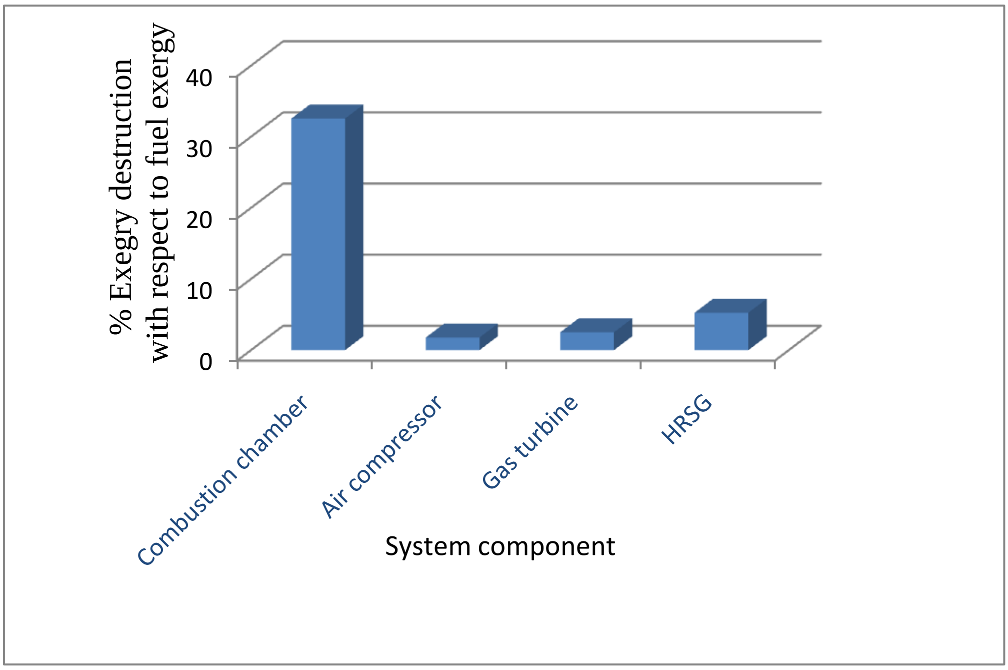

Fig. 4.5 Variation of percentage exergy destruction with respect to fuel exergy for cogeneration system component

In the preliminary study, the accuracy of our developed computer program has been simulated the basic frame 6B simple cycle generation set under ISO conditions (101kPa, 288K,60% RH).The performance data has been taken from “Captive Power Plant of leading petrochemical organization in India”.

4.2.2 Cogeneration power plant at part load

Table 4.9 Input data for analysis of cogeneration plant at part load

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)	8
Isentropic efficiency of compressor (η_{SC}), in %	0.85
Turbine inlet temperature (TIT) or maximum cycle temperature (T_3), in K	1527
Isentropic efficiency of Turbine (η_{GT}), in %	0.86
Exhaust pressure of combustion prod after HRSG (P_5), in bar	1.14
Mass flow rate of steam generated in HRSG (m_{st}) in kg/s	19.44
Pressure of steam generation (P_7) in bar	90
Pressure of feed water at inlet of HRSG (P_6), in bar	90
Temperature of feed water at inlet to HRSG (T_6), in K	383
Net power output of the plant (W_{net}), in MW	20
Injection pressure of fuel (methane) (P_f), in bar	12
Injection temperature of fuel (methane) (T_f), in K	298.15
Pressure drop in HRSG on the gas side, in %	5

Table 4.10 Operating parameters at state points of the cogeneration system at part load

State	Substance	Mass flow rate (kg/s)	Temperature (K)	Pressure (bar)
1	Air	73.61	298.15	1.013
2	Air	73.61	572.4	7.89
3	Combustion Products	75.22	1420	7.12
4	Combustion products	75.22	990.6	1.14
5	Combustion products	75.22	507.9	1.083
6	Water	19.44	383	90
7	Steam	19.44	760.1	90
f	Methane	2.062	311	16

Table 4.11 Calculated parameters at state points of the cogeneration system at part load

State	Substance	Enthalpy (kJ/kmol)	Entropy (kJ/kmolK)	Total exergy (MW)
1	Air	-4713	199.9	0
2	Air	3442	201.3	19.373
3	Combustion Products	-121.3	234.4	73.997
4	Combustion products	-15498	237	30.236
5	Combustion products	-29250	212	6.226
6	Water	8428	210.5	0.647
7	Steam	60598	102.3	8.791
f	Methane	-74875	-	88.739

Table 4.12 Exergy destruction data in each component of gas turbine cogeneration system at part load

Component	Exergy Destruction	Exergy Destruction	Exergy Destruction

	Rate (MW)	(Percentage a)	(Percentage b)
Combustion chamber	27.0772	76.45	30.5
Air compressor	1.648	4.14	1.65
Gas turbine	2.046	5.77	2.3
HRSG	4.67	13.18	5.26
Overall plant	35.4412	100	39.71

Table 4.13 Comparative exergy destruction for each component of gas turbine cogeneration system (part load vs full load)

Ccomponent	Exergy Destruction Rate (MW)		Exergy Destruction (Percentage a)		Exergy Destruction (Percentage b)	
	Part load	Full load	Part load	Full load	Part load	Full load
Combustion chamber	27.077	36.97	76.45	77.31	30.5	32.56
Air compressor	1.648	2.02	4.14	4.22	1.65	1.77
Gas turbine	2.046	2.847	5.77	5.95	2.3	2.5
HRSG	4.67	5.98	13.18	12.5	5.26	5.26
Overall plant	35.441	47.817	100	100	39.71	42.09

It has been calculated that performance parameters the first law efficiency, second law efficiency and electrical to thermal energy ratio of cogeneration system at full load is more than part load. Table 4.13

Table 4.14 Performance parameters for cogeneration system

Parameters	Cogeneration	
	Full load	Part load
First law efficiency (in %)	67	59.2
Second law efficiency (in %)	64.5	56.4
Electrical to thermal energy ratio(in %)	0.49	0.44
Fuel air ratio	.0439	.036
Mass flow rate of dry air	110.4	73.61
Specific Power output		.271
Specific Fuel consumption		
Efficiency of HRSG		
Fuel energy saving ratio		

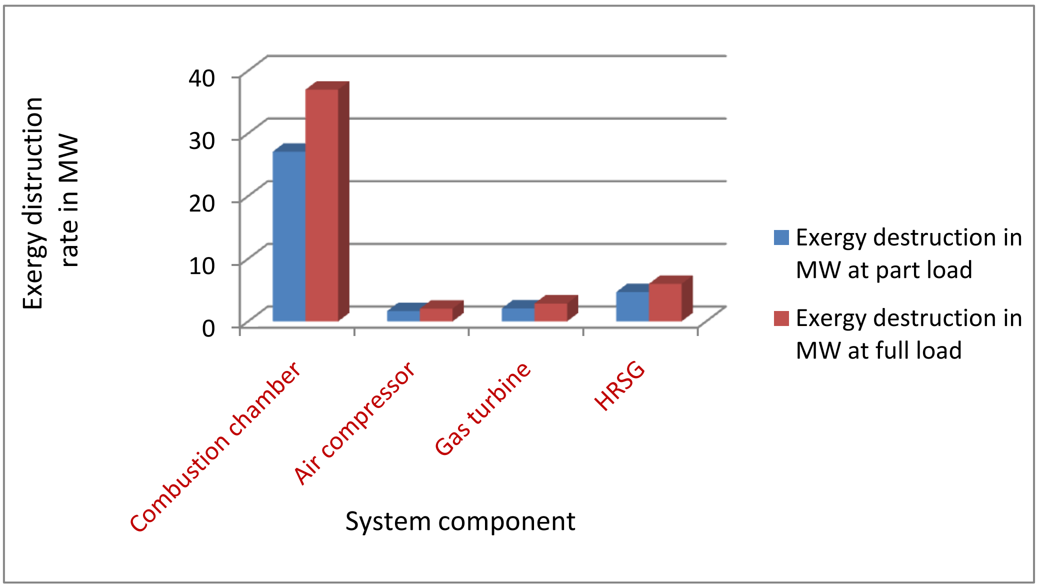

Fig 4.6 Comparison of exergy destruction rate between part load and full load for cogeneration system component in MW

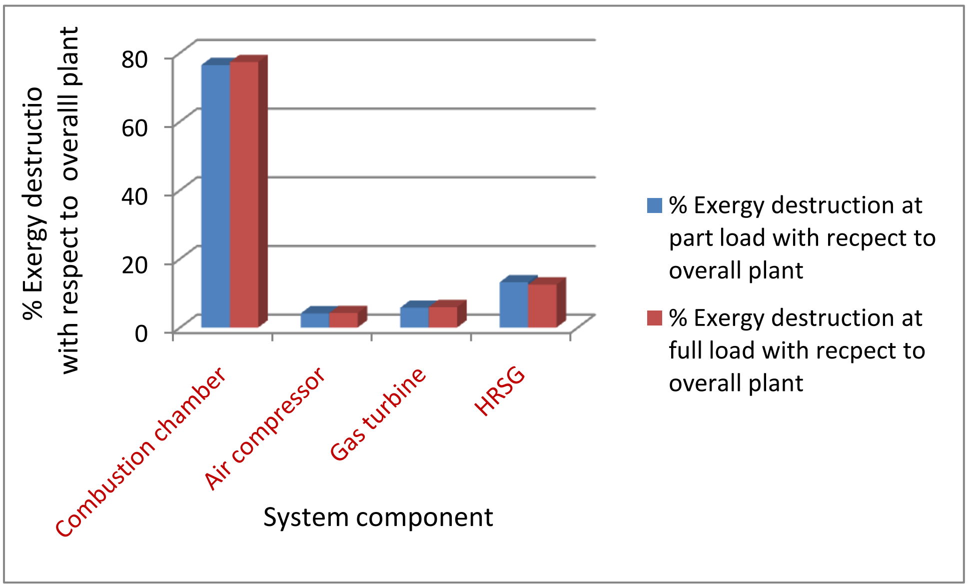

Fig. **4.7** Comparison of percentage exergy destruction with.respect.to overall plant between part load and full load for cogeneration system component

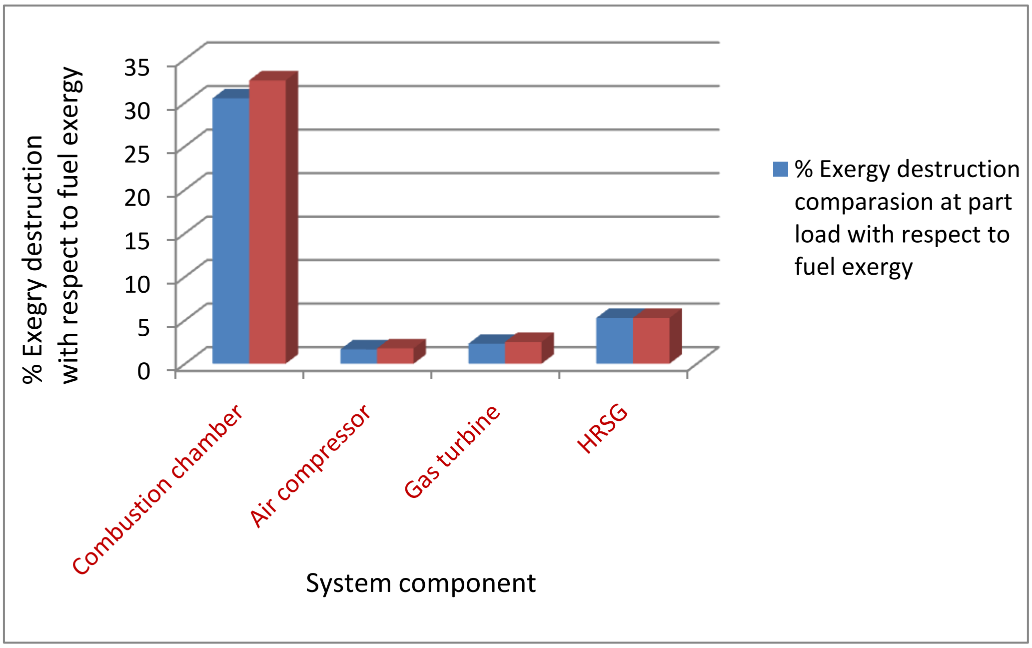

Fig.**4.8** Comparison of percentage exergy destruction with.respect.to fuel exergy between part load and full load for cogeneration system component

It has been seen from above analysis that exergy destruction decreases with reduction in load, but rate of load reduction dominate the reduction in exergy; so that it is not desirable to run the system on part load.

4.2.3 Case –II (Trigeneration power plant)

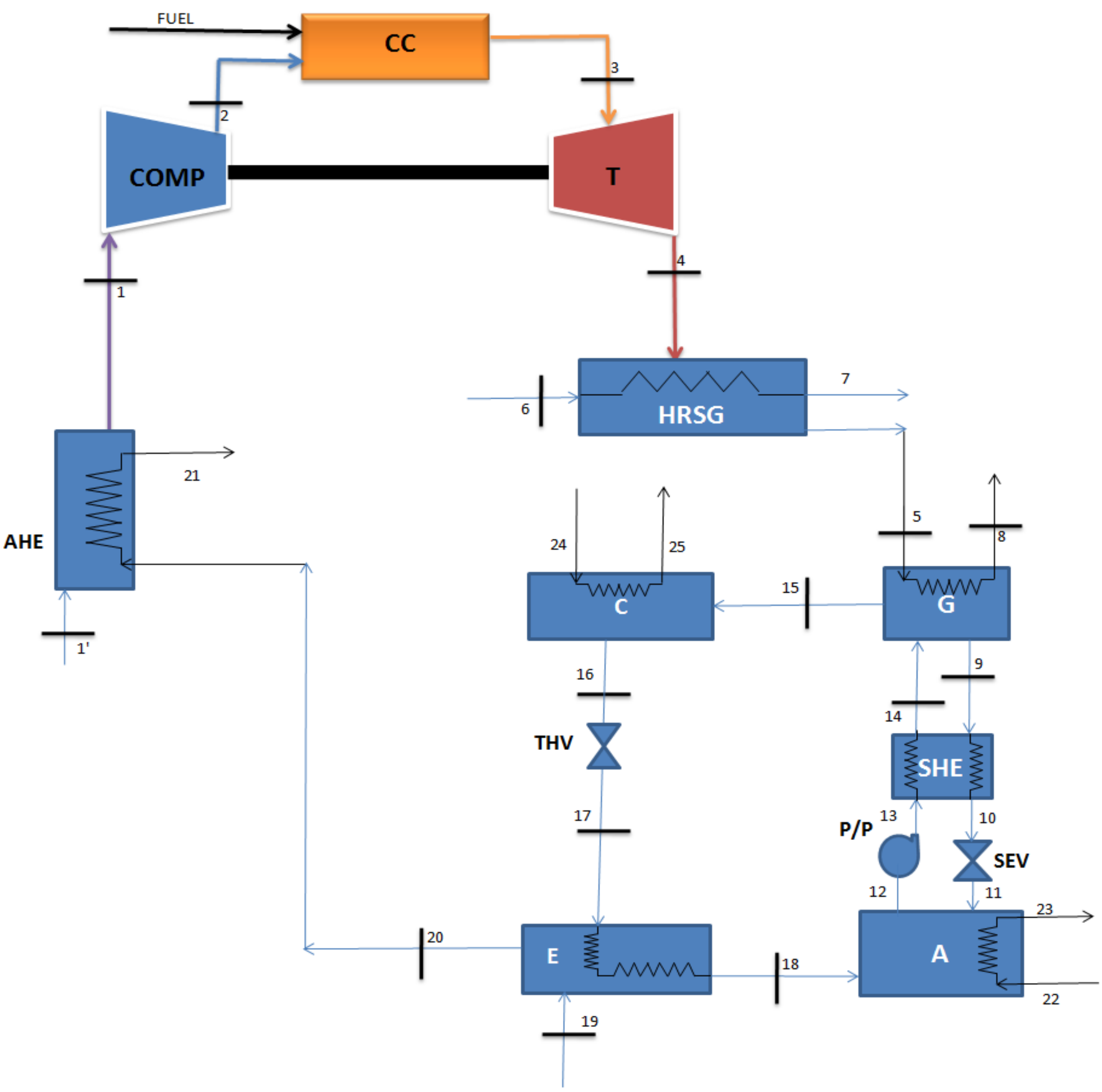

Fig 4.9 Schematic of trigeneration system

Table 4.15 Input data for analysis of trigeneration plant

Air inlet pressure to compressor (P_1), in bar	1.013
Air inlet temperature to compressor, (T_1) in K	283.15
Pressure ratio of compressor (r_p)	10
Isentropic efficiency of compressor (η_{SC}), in %	0.85

Turbine inlet temperature (TIT) in K	1527
Isentropic efficiency of Turbine (η_{GT}), in %	0.86
Exhaust pressure of combustion prod after HRSG (P_5), in bar	1.14
Mass flow rate of steam generated in HRSG (m_{st}) in kg/s	27.77
Pressure of steam generation (P_7) in bar	90
Pressure of feed water at inlet of HRSG (P_6), in bar	90
Temperature of feed water at inlet to HRSG (T_6), in K	383
Net power output of the plant (W_{net}), in MW	40
Injection pressure of fuel (methane) (P_f), in bar	22
Injection temperature of fuel (methane) (T_f), in K	298.15
Pressure drop in HRSG on the gas side, in %	5
Generator temperature in ˚C	90
Evaporator temperature in ˚C	5
Condenser temperature in ˚C	35
Absorber temperature in ˚C	35
Effectiveness of solution heat exchanger	0.85
Inlet temperature of cooling media at absorber in ˚C	15
Outlet temperature of cooling media at absorber in ˚C	35
Inlet temperature of cooling media at condenser in ˚C	25
Outlet temperature of cooling media at condenser in ˚C	35
Inlet temperature of cooling media at evaporator in ˚C	15
Outlet temperature of cooling media a evaporator in ˚C	10
Pressure drop in generator on the gas side, in %	5

Table 4.16 Operating parameters at state points of the trigeneration system

State	Substance	Mass flow rate (kg/s)	Temperature (K)	Pressure (bar)
1	Air	85.65	298.15	1.013
2	Air	85.65	611.34	10.5
3	Combustion Products	87.71	1527	9.61
4	Combustion products	87.71	1005.5	1.099
5	Combustion products	87.71	507.9	1.213
6	Water	19.44	383	90
7	Steam	19.44	760.1	90
8	Combustion products	87.71	210	1.213
9	Weak solution	65.07	90	5.625
10	Weak solution	65.07	43.25	5.625
11	Weak solution	65.07	43.25	0.87
12	Strong solution	55.7	35	.87
13	Strong solution	55.7	35	5.624
14	Refrigerant	0.121	69.92	5.624
15	Refrigerant	0.121	90	5.624
16	Refrigerant	0.121	35	5.625
17	Refrigerant	0.121	5	0.87
18	Refrigerant	0.121	5	0.87
19	Cooling water	13.66	20	1.013
21	Cooling water	13.66	10	1.013
22	Cooling water	13.66	25	1.013
23	Cooling water	18.92	35	1.013
24	Cooling water	7.924	25	1.013
25	Cooling water	7.924	35	1.013
f	Methane	2.062	311	16

Table 4.17 Calculated parameters at state points of trigeneration system

State	Substance	Enthalpy (kJ/kmol)	Entropy (kJ/kmolK)	Total exergy (MW)
1	Air	-4713	199.9	0
2	Air	4632	201.3	27.619
3	Combustion product	694.5	235.2	103.978
4	Combustion product	-17309	238.4	41.299
5	Combustion product	-32258	217	9.427
6	Water	8428	210.5	0.647
7	Steam	60598	102.3	8.791
8	Combustion product	-37975	207.7	110.45
9	Weak solution	235.6	0.4665	22.587
10	Weak solution	152	0.4665	25.553
11	Weak solution	152	0.287	25.553
12	Strong solution	85.88	0.2207	29.791
13	Strong solution	85.88	0.2207	29.791
14	Refrigerant	157.5	0.4365	29.823
15	Refrigerant	48074	156	7.118
16	Refrigerant	48074	9.097	4.428
17	Refrigerant	2641	9.506	4.428
18	Refrigerant	45213	162.6	6.721
19	Cooling water	1135	4.039	49.0123
20	Cooling water	758.2	2.7	48.7749
21	Cooling water	2266	6.61	5.732
22	Cooling water	1889	6.61	26.4797
23	Cooling water	2642	9.096	26.7507
24	Cooling water	1889	6.61	26.4287
25	Cooling water	2642	9.096	26.6992
f	Methane	-74875	-	111.657

Table 4.18 Exergy destruction data of trigeneration system

Component	Exergy Destruction Rate (MW)	Exergy Destruction (Percentage a)	Exergy Destruction (Percentage b)
Combustion chamber	36.970	75.2890	33.11
Air compressor	2.020	4.1137	1.80
Gas turbine	2.847	5.7978	2.54
HRSG	5.98	12.1782	10.90
VAR System	1.2871	2.6211	5.36
Overall plant	49.1041	100	53.71

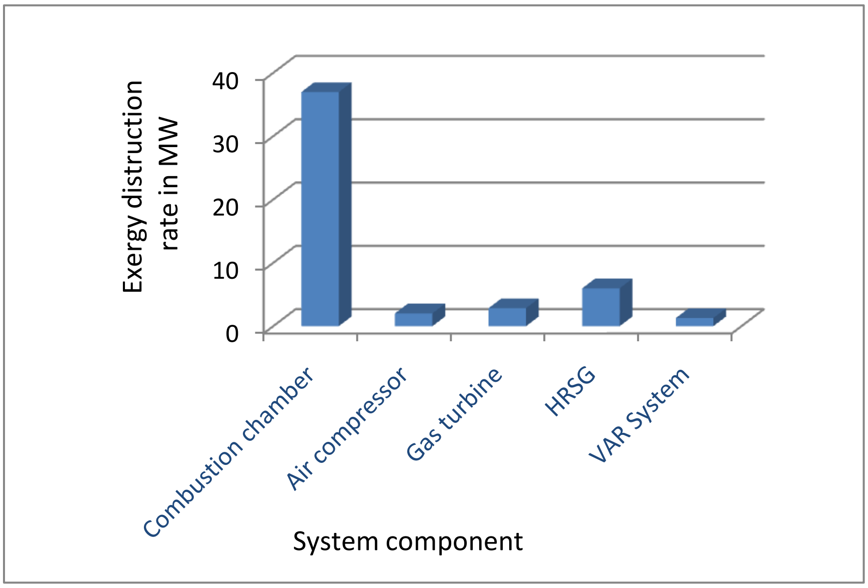

Fig. 4.10 Variation of exergy destruction for trigeneration system Component

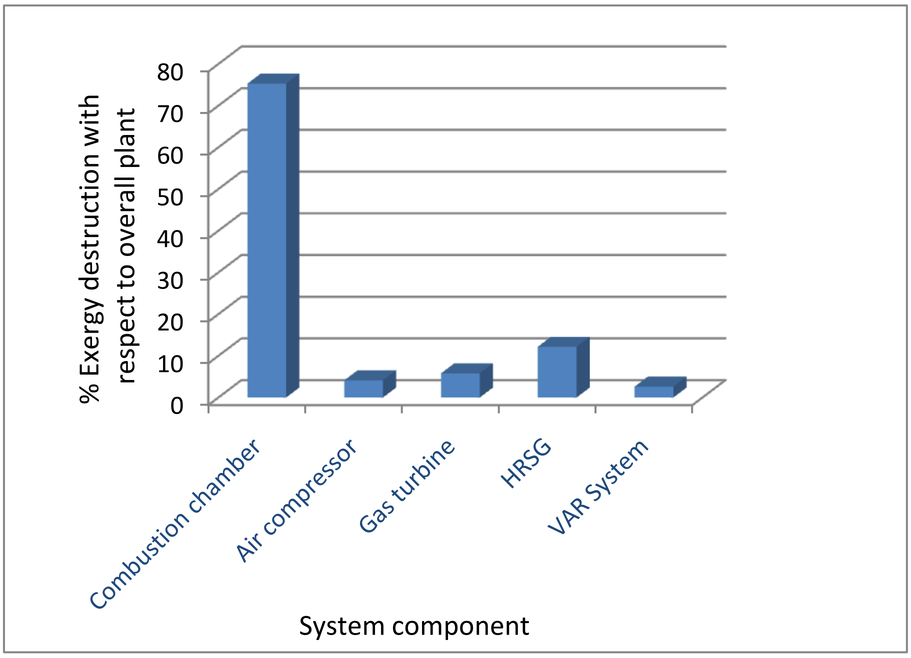

Fig. 4.11 Variation of percentage exergy destruction with respect to Overall plant for trigeneration system

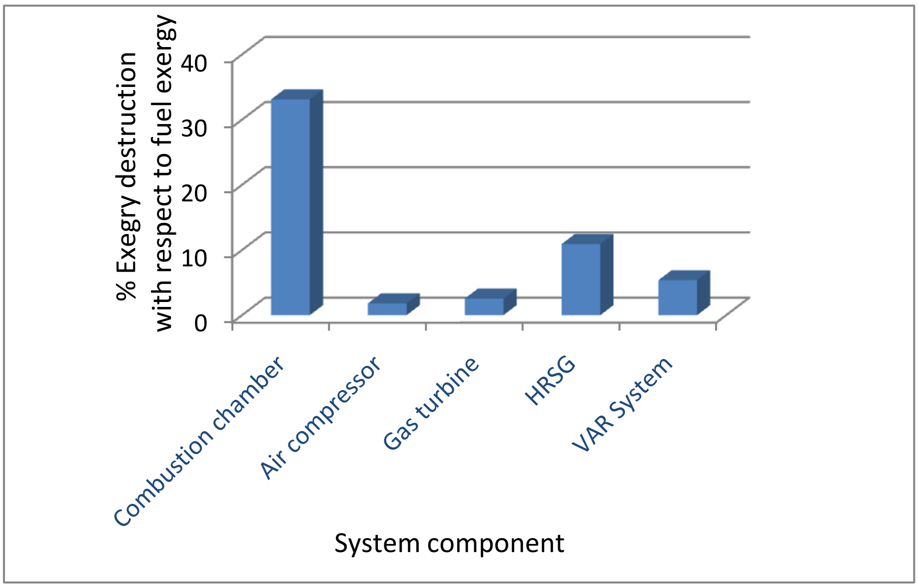

Fig. 4.12 Variation of percentage exergy destruction with respect to Fuel exergy for trigeneration system

Table 4.19 Exergy destruction data VAR system

Component	Exergy Destruction Rate (MW)	Exergy Destruction (Percentage a)	Exergy Destruction (Percentage b)
Generator	0.9247	71.84	17.51
Solution heat exchanger	0.001834	0.0014	0.03
Absorber	0.2236	17.37	3.7
Evaporator	0.07476	5.8	1.25
Condenser	0.05459	4.24	0.91
Expansion valve	0.00762	0.00592	0.12
	1.287104	100	23.52

Table 4.20 Performance parameter for trigeneration system

Parameters	Trigeneration
First law efficiency (in %)	75.6
Second law efficiency (in %)	73.4
Electrical to thermal energy ratio(in %)	0.4

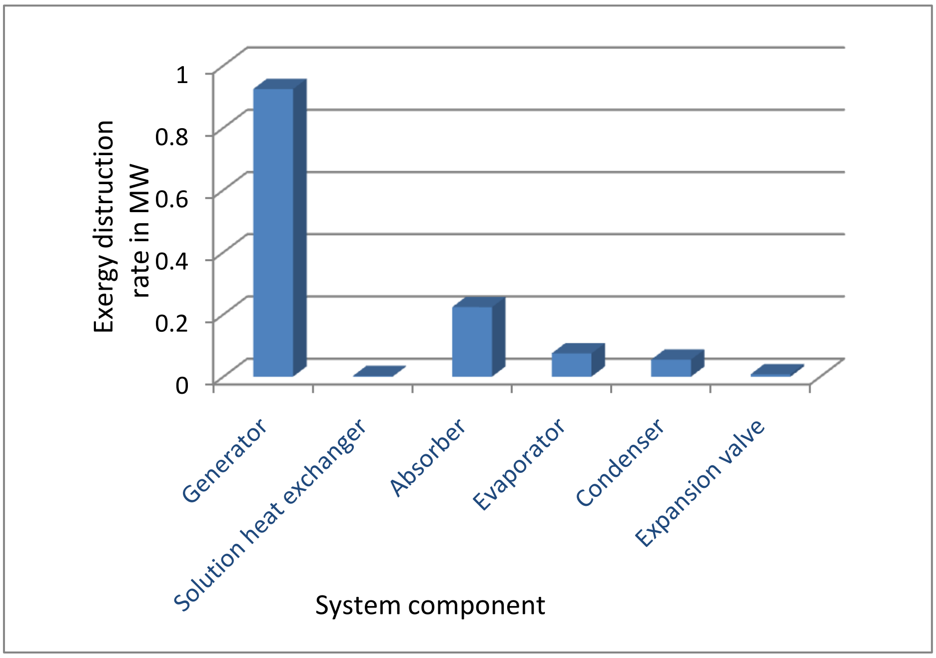

Fig.4.13 Variation of exergy destruction for VAR system Component

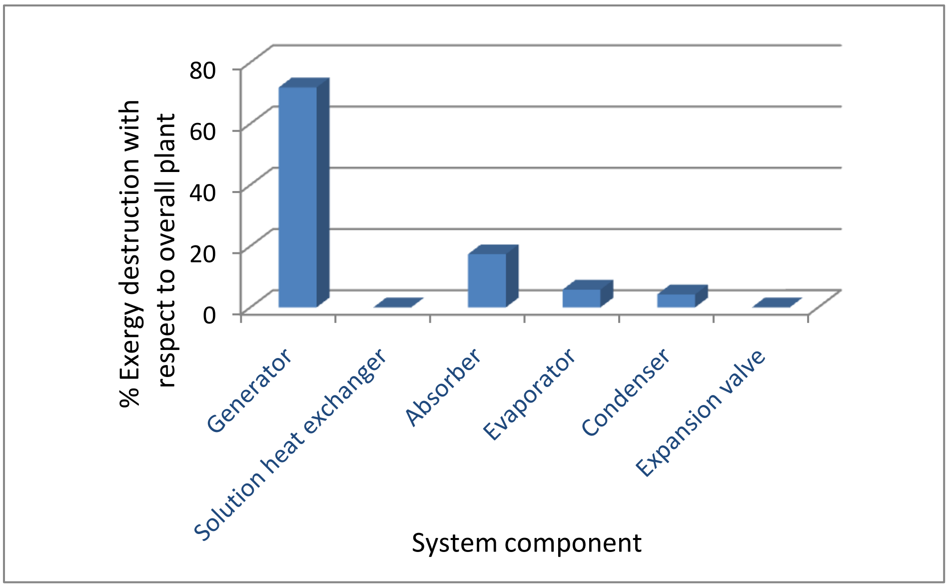

Fig. 4.14 Variation of percentage exergy destruction with respect to Overall plant for VAR system

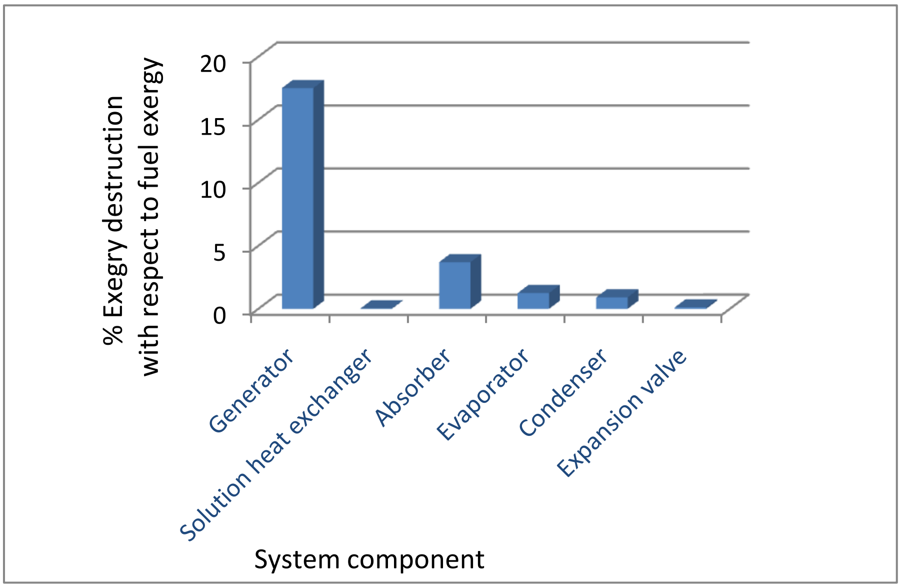

Fig. 4.15 Variation of percentage exergy destruction with respect to Fuel exergy for VAR system

In the preliminary study, the accuracy of our developed computer program has been simulated by the basic LM 6000 simple cycle generation set under ISO conditions (101kPa, 283K, 60% RH).

From the above analysis it has been observed that the exergy destruction in trigeneration plant is comparable with cogeneration plant, but trigeneration having additional advantage of inlet air cooling with minor exergy destruction in VAR system i.e. 2.6211% of total exergy destruction.

The performance has been taken from "Captive Power Plant of leading petrochemical organization in India".

It has been calculated that performance parameters the first law efficiency, second law efficiency and electrical to thermal energy ratio of cogeneration system at full load is more than part load. On the other hand in trigeneration system first and second law efficiency is more than cogeneration in both mode, but electrical to thermal energy ratio is less as compared to cogeneration system. The typical value of the above discussed parameters is given in Table 4.21

Table 4.21 Performance parameter for cogeneration and trigeneration system

Parameters	Cogeneration		Trigeneration
	Full load	Part load	
First law efficiency (in %)	67	59.2	75.6
Second law efficiency (in %)	64.5	56.4	73.4
Electrical to thermal energy ratio(in %)	0.29	0.34	0.44

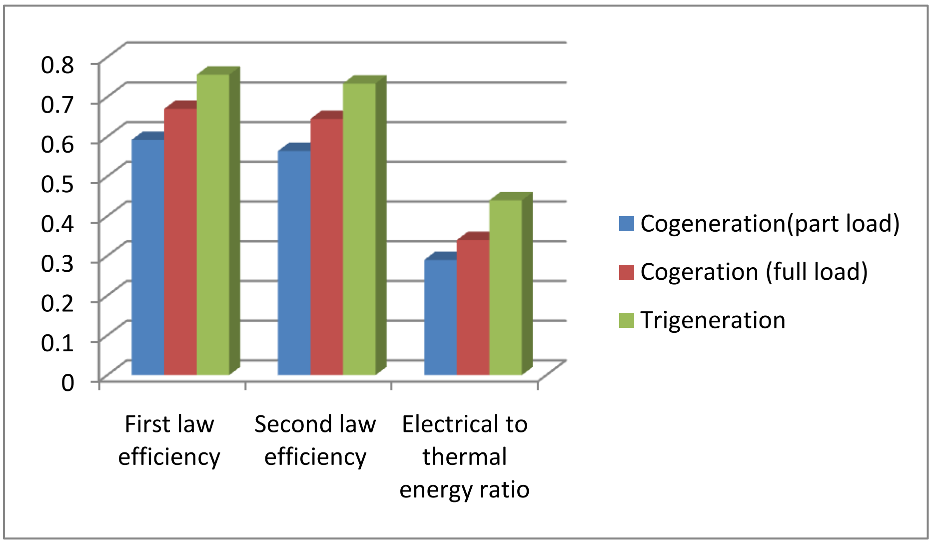

Fig. 4.16 comparison of performance parameter

4.3 Parametric studies

The effect of pressure ratio (r_p) across the compressor, turbine inlet temperature (TIT), ambient temperature and inlet air temperature on the first law efficiency (η_I) and power-to-heat ratio is obtained by the energy balance approach or the first law analysis of the cycle. However, the exergy destruction or thermodynamic losses of each component and second law efficiency (η_{II}) of the cycle has also been investigated under the exergy balance approach or second-law analysis of the cycle that has been discussed in chapter 3. To study the effect of these operating variables on the performance parameter of the system, operating under different conditions, the following common characteristics and their corresponding ranges were chosen:

1. **Compressor Pressure ratio (r_p)**: The compressor pressure ratio has been varied for a range of 4 to 24.

2. **Turbine Inlet temperature (TIT):** Due to metallic advancements, the value of TIT is generally increasing. In the present simulation work, the range selected for TIT is 1300 to 1700 ˚C.

3. **Characteristics of fuel:** Natural gas is taken as fuel having the lower heating value of 802361kJ/kmole.

4. **System component losses :**

   Pressure drop in combustion chamber = 5%

   Pressure loss in HRSG gas side = 4%

   Pressure loss in generator of VARS = 5%

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

   Compressor isentropic efficiency η_c = 85%

   Gas Turbine isentropic efficiency η_T = 90%

   Efficiency of combustion chamber η_{cc} =95%

   Generator efficiency η_g = 98%

   Effectiveness of condenser = 85%

   Effectiveness of evaporator=85%

   Effectiveness of generator = 80%

Effect of Pressure Ratio

As pressure ratio increases the air temperature at the inlet of combustion chamber increases which results in decreasing the heat added to the cycle. The ratio of net work output to heat added represent the first law efficiency, hence as pressure ratio increases the first law efficiency of trigeneration system increases as shown in Figure 4. Figure also shows the variation of second law efficiency which is more accurate measure of thermodynamic performance, since the quality of fuel i.e. exergy associated with heat addition is higher than heating value or energy of fuel, the exergy of the fuel would increases while bringing it from ambient pressure to combustion pressure at ambient temperature. Hence exergy associated with the heat addition will be equal to exergy associated with the heat addition with the heating value of fuel plus exergy increase, therefore the second law efficiency is slightly lower the first law efficiency of trigeneration system. It is shown in figure 4.17

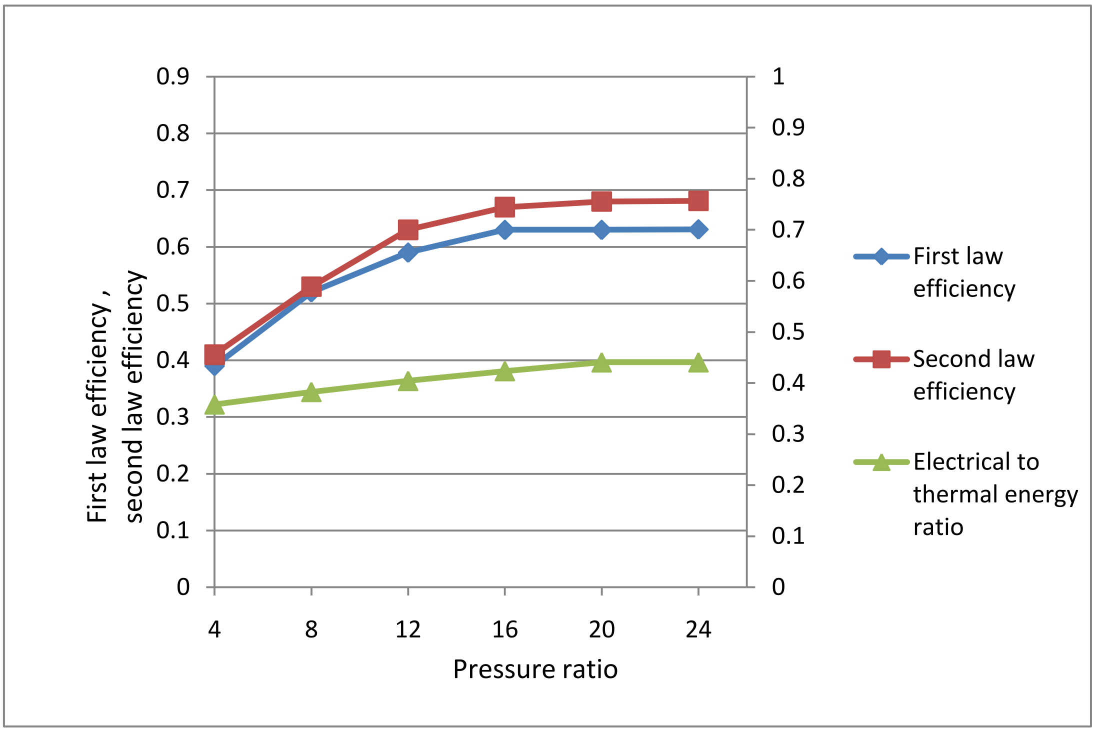

Fig.4.17 Effect of variation of pressure ratio on first law efficiency, second law efficiency and electrical to thermal ratio

Effect of variation of Turbine inlet air temperature

Fig 4.18 shows the variation of first law efficiency (η_I), second law efficiency (η_{II}) and electrical to thermal ratio (R_{ET}) with respect to change in TIT for ($r_{p=}$ 10, $T_{e\,=}$ 5˚C).It is found that the first law efficiency increases with increase in TIT.This is because increasing TIT leads to significant increase in net work output and insignificant increase in heat addition of cycle. Therefore first law efficiency increases with increase in TIT

It is also shows that the variation the variation of second law efficiency is slightly lower than first law efficiency. This difference between second law efficiency and first law efficiency for trigeneration occurs due to exergy associated with cold is much less than the energy of process heat. it is also seen from Figure 4.18 that the turbine inlet temperature does not have significant effect on electrical to thermal energy ratio for trigeneration cycle because increase in TIT causes significant increase in power output, process heat and cold and the improvement in process heat and cold.

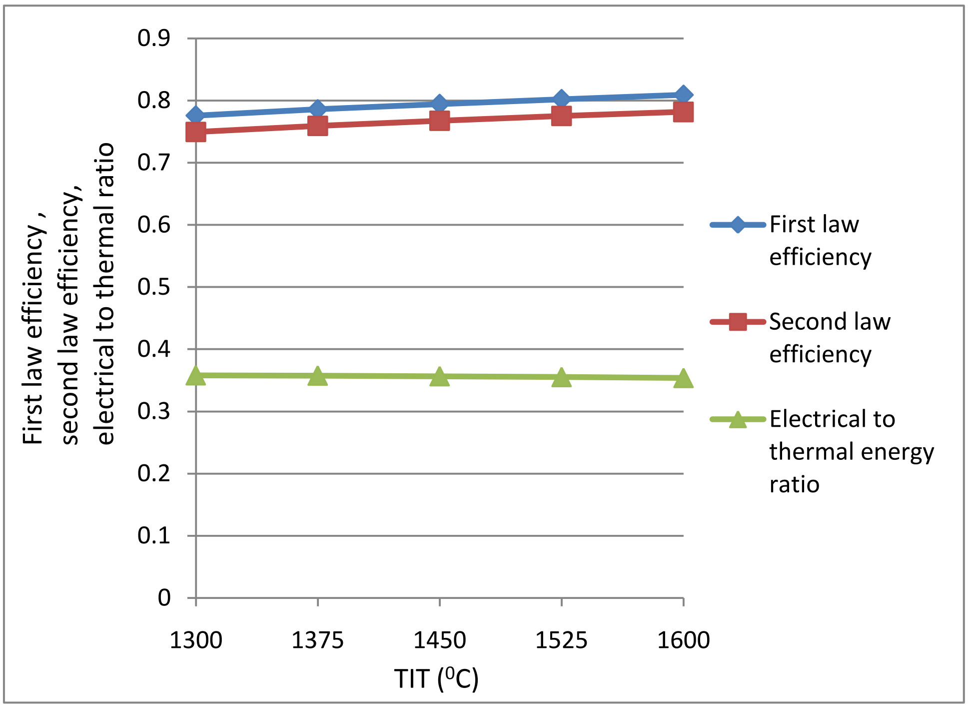

Fig. 4.18 Effect of variation of Turbine inlet temperature (TIT) on first law efficiency, second law efficiency and electrical to thermal energy ratio

Effect of variation of Turbine evaporator temperature

As shown in figure 4.19 the first law efficiency of trigeneration system slightly increases with the increase in increase in evaporator temperature. This is because at higher evaporator temperature ,the cooling load will higher .The second law efficiency also increases with evaporator temperature but the magnitude of increase is very small as the exergy associated with the cold is very small. The electrical to thermal ratio decreases with increase in evaporator temperature due to increase in cold.

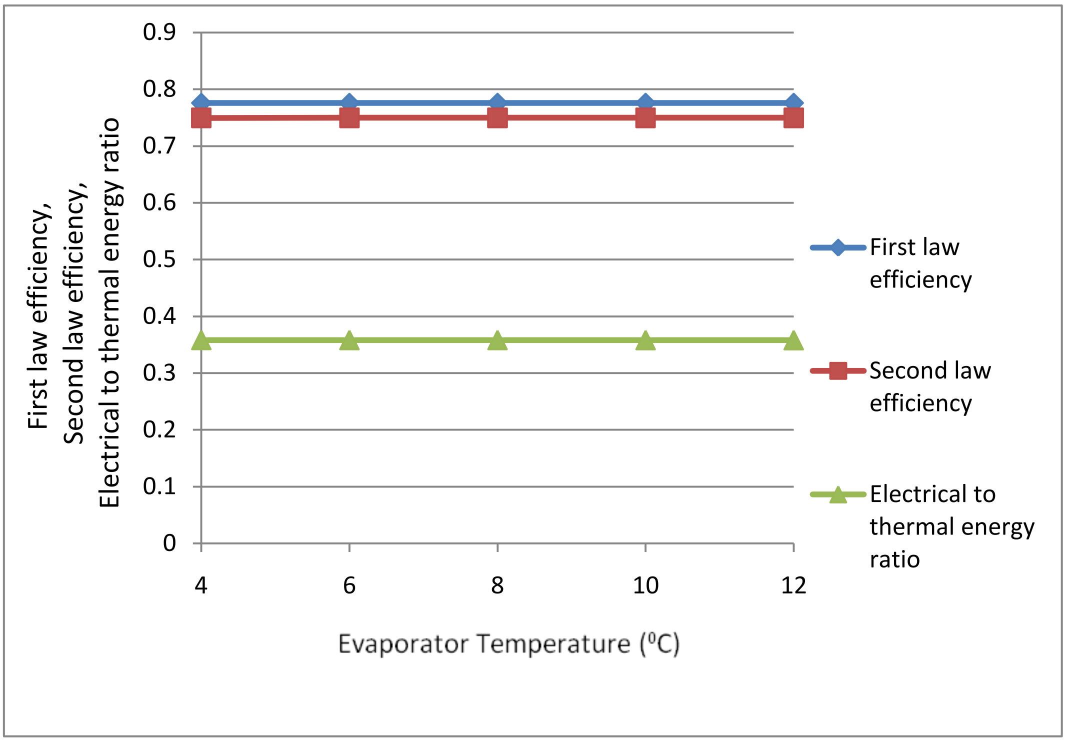

Fig. 4.19 Effect of variation of evaporator temperature on first law efficiency, second law efficiency and electrical to thermal ratio

Effect of Ambient temperature

Figure 4.20 shows the variation of power output with ambient temperature. As air temperature rises, its density falls. Thus, the volumetric flow rate remains constant; the mass flow rate is reduced as air temperature rises. Power output is also reduced as air temperature rises because power output is proportional to mass flow rate. As per ISO condition optimum power output occurs at 15˚C that is shown in figure 4.21

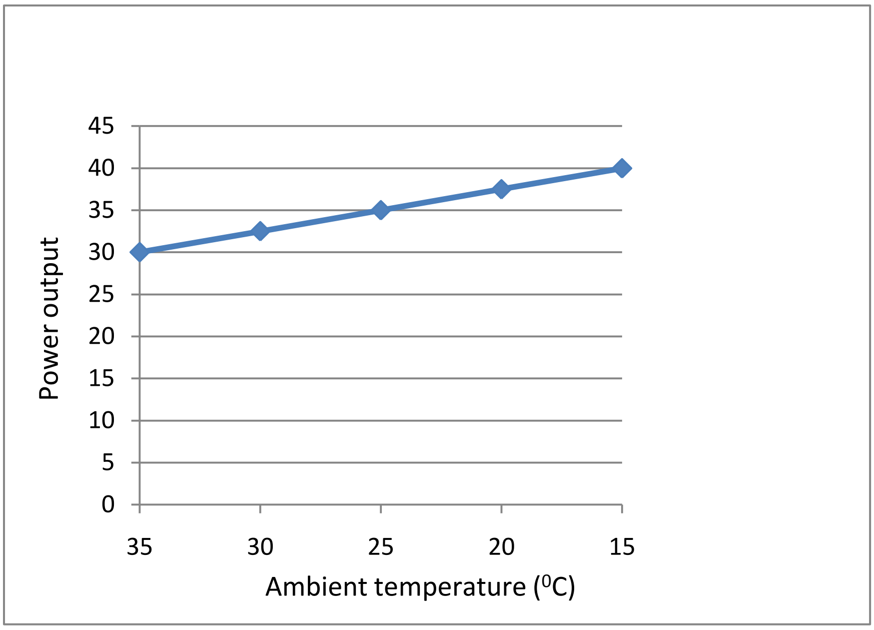

Fig.4.20 Effect of variation of ambient temperature on power output

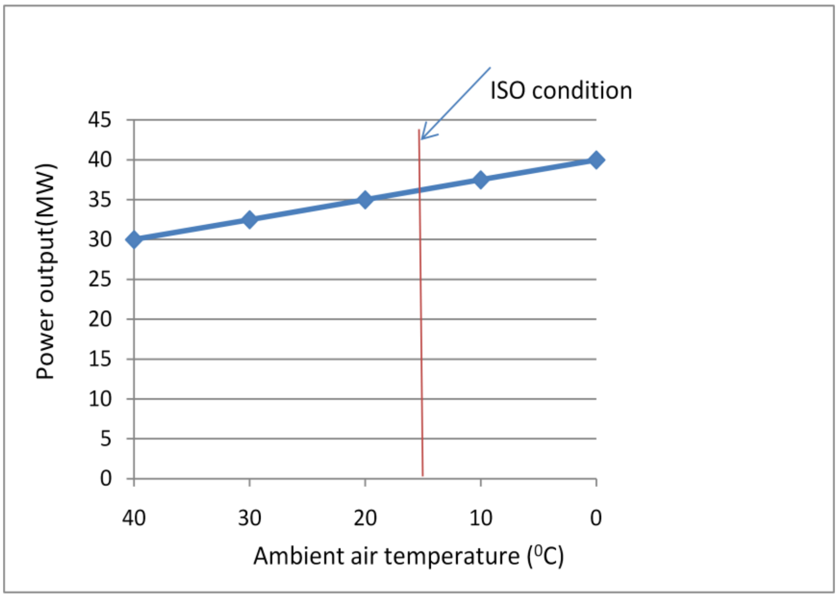

Fig.4.21 Effect of variation of ambient temperature on power output with respect to ISO condition

Effect of Pressure ratio on Exergy Destruction

Figure 4.22 shows the variation of magnitude of exergy destruction in each component of system with change in pressure ratio. It is found the exergy destruction in combustion process dominates it represent over 70% of the total exergy destruction in overall system. As the pressure ratio increases the exergy destruction in the combustion chamber decreases significantly. this is because of the increase in pressure ratio implies lower difference of exergy between combustion products and compressed air but its difference with exergy carried by fuel drops. It is further shown in Table 4.22 that as the pressure ratio increases the exergy destruction in HRSG decreases. this is because of the higher pressure ratio results in higher exergy of combustion products and lower turbine exhaust exergy which leads to the higher turbine output.

Table.4.22 Effect of variation of pressure ratio on exergy destruction in different components of trigeneration system for TIT=1400˚C

prc	$E_{D,CC}$ (kW)	$E_{D,C}$ (kW)	$E_{D,HRSG}$ (kW)	$E_{D,T}$ (kW)	$E_{D,VARS}$ (kW)
4	85698	2577	5564	2721	1798
6.222	62483	2568	4327	3151	1675
8.444	53154	2634	3965	3540	1686
10.67	48003	2720	3859	3903	1732
12.89	44707	2814	2853	4247	1792
15.11	42412	2911	2133	4581	1858
17.33	40727	3011	1987	4907	1927
19.56	39444	3113	1901	5228	1999
21.78	38444	3216	1881	5548	2072
24	37651	3321	1671	5867	2147

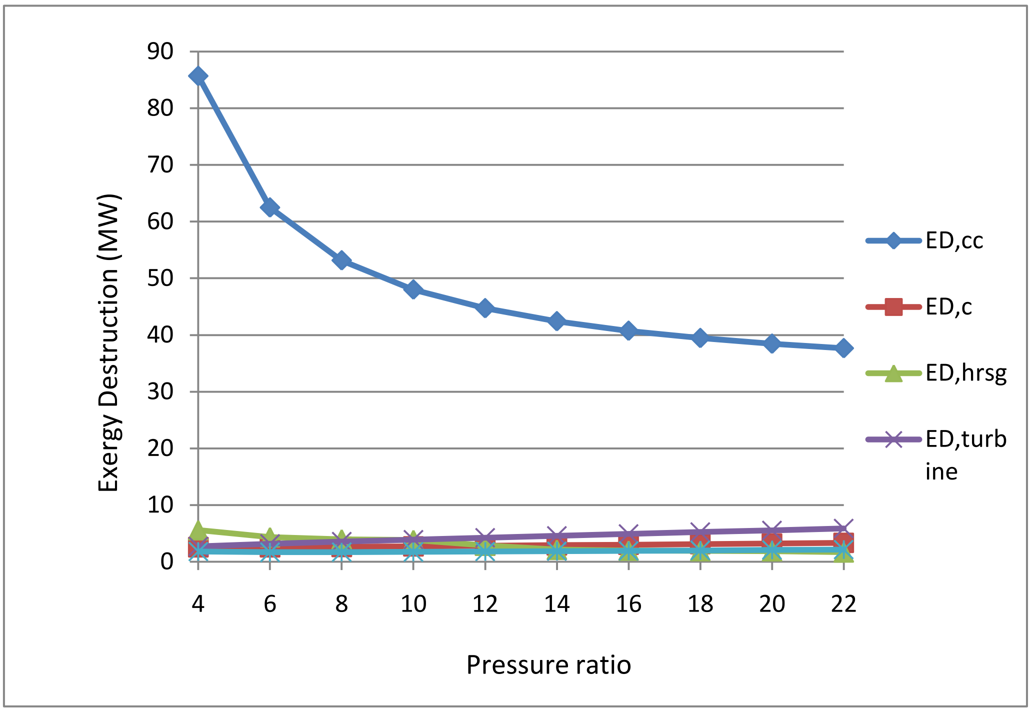

Fig 4.22 Variation of exergy destruction (MW) for trigeneration system Components with respect pressure ratio

Effect of Pressure ratio on Exergy Destruction

Figure 4.23 shows the variation of magnitude of exergy destruction in each component of system with change in turbine inlet temperature. As TIT increases the exergy destruction in the combustion chamber increases, because the mean temperature of heat addition increases. The exergy destruction in HRSG increases because the temperature difference between two heat exchanging fluid (flue gases and water/steam) increasesfor the given pressure ratio of the cycle, more process heat is produced due to more steam generated by HRSG at higher TIT

Table 4.23 Effect of variation of TIT on exergy destruction in different components of trigeneration system for prc=10

TIT (°C)	$E_{D,CC}$ (kW)	$E_{D,C}$ (kW)	$E_{D,HRSG}$ (kW)	$E_{D,T}$ (kW)	$E_{D,VARS}$ (kW)
1300	53075	2441	3234	812	1360
1356	51669	2278	3022	987	1049
1411	50390	2134	2835	1024	757.9
1467	49220	2007	2669	1165	482.7
1522	48143	1892	2521	1346	220.1
1578	47146	1789	2388	1496	202
1633	46220	1696	2267	1765	182
1689	45354	1612	2158	1897	145
1744	44542	1534	2058	2087	132
1800	43777	1463	1967	2341	109

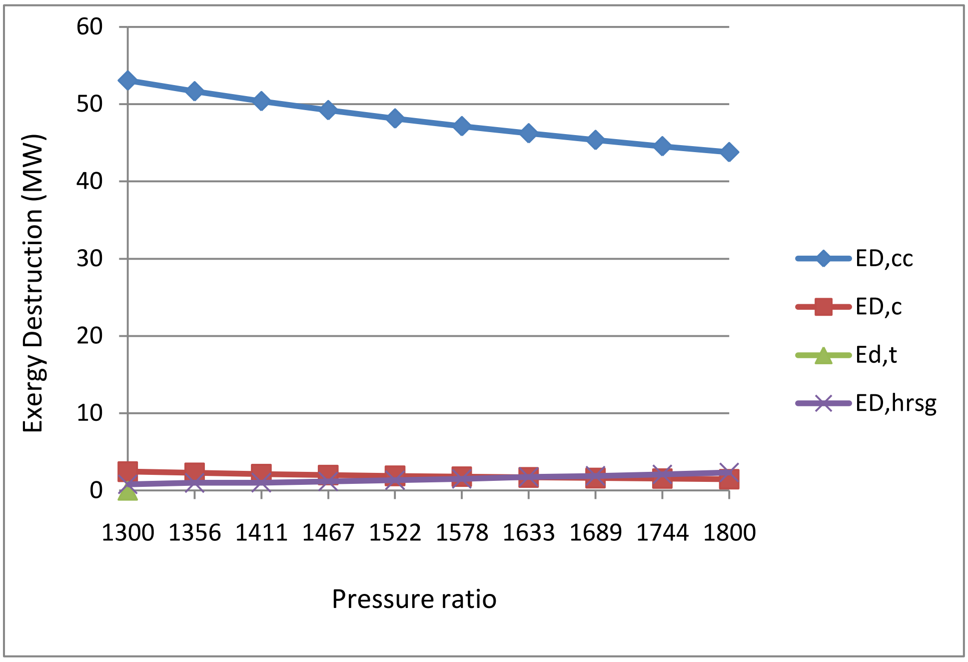

Fig 4.23 Variation of exergy destruction (MW) for trigeneration system Components with respect TIT

5.1 Conclusions

The energy and exergy balance equations, which are applicable to any thermal system, have been applied to the trigeneration system for combined production of power, heat and refrigeration. From thermodynamic point of view, the combination of gas turbine with absorption chilling machine in this trigeneration system proves to be highly efficient, because the flue gas from heat recovery steam generator is used as a heat source for vapour absorption refrigeration as described in this study.

The values of performance parameters for cogeneration and trigeneration system

Parameters	**Cogeneration**		**Trigeneration**
	Full load	Part load	
First law efficiency (in %)	67	59.2	75.6
Second law efficiency (in %)	64.5	56.4	73.4
Electrical to thermal energy ratio	0.29	0.34	0.44

The trigeneration system is more efficient as compared to cogeneration. The conclusion of parametric study for trigeneration as follow

- Maximum energy is destroyed during the combustion and steam generation process; it represents over 82% of the total energy destruction in the overall system.
- The first law efficiency, second law efficiency and electrical to thermal energy of Trigeneration increases with the increase in pressure ratio up to 16, after that it almost remains constant.
- The first law efficiency, electrical to thermal energy ratio, and the second law efficiency Trigeneration system increases with the increase in turbine inlet temperature.

- The energy destruction in combustion chamber and heat recovery steam generator decreases significantly with the in pressure ratio but increases significantly with the increase in turbine inlet temperature.

- The energy destruction in the generator, the absorber, and the condenser increases slightly with the evaporator temperature, while it decreases in the throttling valve, the evaporator and the heat exchanger solution.
- The first law efficiency and second law efficiency for Trigeneration is found to be almost constant with the variation of evaporator temperature but the electrical to thermal energy ratio for Trigeneration decreases slightly with the increase in evaporator temperature.
- At a given TIT, process heat pressure, and evaporator temperature, the energy destruction in compressor and turbine increases with the increase in pressure ratio.

5.2 Recommendations of future work

The present computer program can be easily extended in following areas-

1. The present Trigeneration system can be extended by incorporating reheat, steam injection technology for parametric studies by using of double effect vapour absorption refrigeration system.
2. The present study can be further extended for Techno-economic analysis.

Chapter 5

CONCLUSIONS AND FUTURE SCOPE OF WORK

5.1 Conclusions

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