STUDY OF GAS TURBINE COGENERATION CYCLE WITH INLET EVAPORATIVE COOLING AND REHEATING

A DISSERTATION

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR THE AWARD OF DEGREE
OF

MASTER OF TECHNOLOGY
IN
THERMAL ENGINEERING

Submitted by:

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CANDIDATE'S DECLARATION

I, Jyoti Rautela, Roll No. 2K18/THE/18 student of M.Tech (Thermal Engineering), hereby declare that the project Major project II titled **"STUDY OF GAS TURBINE COGENERATION CYCLE WITH INLET EVAPORATIVE COOLING AND REHEATING"** which is submitted by me to the Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfillment of the requirement for the award of the degree of Master of Technology, is original and not copied from any source without proper citation. This work has not previously formed the basis for the award of any Degree, Diploma, Associateship, Fellowship or other similar title or recognition.

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CERTIFICATE

I hereby certify that Major project II titled **"Study of Gas turbine cogeneration cycle with inlet evaporative cooling and reheating"** which is submitted by Jyoti Rautela, 2K18/THE/18 Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfillment of the requirement for the award of the degree of Master of Technology, is a record of the project work carried out by the students under my supervision. To the best of my knowledge this work has not been submitted in part or full for any Degree or Diploma to this University or elsewhere

Place: Delhi | Sh. P.V. Ram Kumar

Date: | SUPERVISOR

Associate Professor

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I also reveal my thanks to all my classmates and my family for constant support.

JYOTI RAUTELA

ABSTRACT

Energy has been a major concern over the years all over the world. Cogeneration is being coined as the efficient utilization of energy in various power plants. Gas turbine being the compact and one of the most reliable means for the generation of power is pretty good to be used for cogeneration. Cogeneration in general is a term used for producing heat and work simultaneously. The exhaust heat from the turbine is used as the process steam for either district heating, cooling at various applications. Cogeneration with gas turbines is the frequently and widely used cogeneration system because this type of system takes the advantage of the relative merit of engine and enhances thermal efficiency at full load conditions as well as part load conditions. Also, it is highly accepted in power industries.

The work performed provides a review of research investigation on cogeneration systems which are carried out during the last few years. Along with the literature review, the performance of various output parameters of gas turbine cogeneration system with reheat and inlet evaporative cooling (proposed cycle) are analyzed parametrically taking compressor pressure ratio, turbine inlet temperature, pinch point temperature and inlet temperature of air as the main variables. The effect of these parameters on power/specific work, efficiency, power to heat ratio, First law efficiency and second law efficiency were plotted with the help of EES software.

Keywords: Gas turbine cycle, Cogeneration, Pinch point temperature, Energy destruction

CONTENTS

LIST OF FIGURES AND TABLES

SYMBOLS USED:

Φ	-	Relative humidity
TIT	-	Turbine Inlet Temperature
η_I	-	First law efficiency or Thermal Efficiency
η_{II}	-	Second law efficiency
CHP	-	Combined heat and power
HRSG	-	Heat recovery steam generator
COP	-	Coefficient Of Performance
Q_f	-	Fuel input
W_{el}	-	Electrical work
$\dot{B}_p$	-	Exergy content of the process heat
$\dot{B}_F$	-	Exergy content of fuel input
Q_P	-	Process heat
$\dot{m}_a$	-	Mass flow rate of air
m_f	-	Mass flow rate of fuel
$\dot{m}_w$	-	Mass flow rate of water
LHV	-	Lower heat value
h	-	Enthapy at a given state
A/F	-	Air fuel ratio
ω	-	Specific Humidity
T_{db}	-	Dry Bulb Temperature
T_w	-	Wet Bulb Temperature
$e_{f_{cc}}$	-	Exergy associated with the fuel in combustion chamber
$e_{f_{reh}}$	-	Exergy associated with the fuel in reheater
W_T	-	Work done by turbine
W_c	-	Work done by compressor
W_{NET}	-	Net work done

CHAPTER 1

INTRODUCTION

1.1 NEED FOR COGENERATION

Major source of electricity supply in India are thermal power plants. The conventional method adopted for generating the power and then supplying it to the customer is wasted in a way that mere one third of the primary energy that is put into the power plant is available to the user in the form of electrical energy in actual scenario (Figure 1.1).The conventional power plant are having an efficiency of 35% only and remaining 65% of energy is lost in the process. There are many causes for this loss, the rejected heat to the air or water surrounding it due to the inherited constraints to the efficiency of the various thermodynamic cycles used in power generation being the major reason for the loss in the process of conversion. Electricity transmission and distribution in the electrical grid contributes to about 10–15% losses [1].

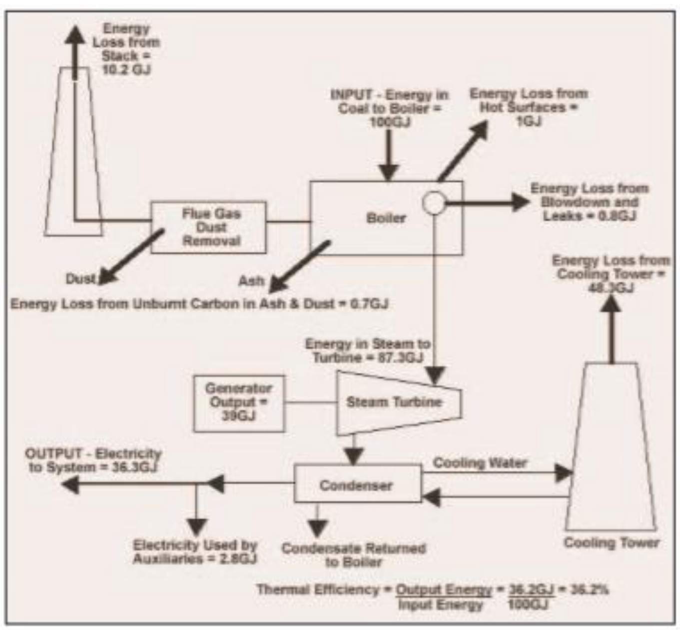

Fig 1.1 Balance in typical coal fired power station [1]

In the last few decades, many attempts have been made to introduce arrangements so as to enhance the performance of power generating thermal systems. Cogeneration is one of the choices for efficient utilization of energy. Cogeneration is coined to describe simultaneous production of electrical and heat energy [2].

1.2 COGENERATION PRINCIPLE:

Cogeneration is production of combined heat and power. The name suggests itself that cogeneration is based on the concept of producing two forms of energy by utilizing only one fuel source. The two forms of energy are thermal energy in the form of heat and other one is either mechanical energy in the form of mechanical work converted into electricity. Cogeneration with gas turbines is the most widely used because the simple cycle gas turbine engine is known for comparatively lower capital cost, higher flexibility and higher reliability without much complexity [3].

Combined Heat and Power (CHP) system or cogeneration system is defined as the generation of two forms of useful energy in a sequential manner from a single primary energy source, namely mechanical work from turbine and heat i.e. thermal energy for district heating and cooling. Mechanical energy can be used either to drive rotating equipment such as compressor, motor or an alternator for producing electricity, fan for numerous services. Heat energy can be utilized in either directly for various process applications or indirect production of steam, hot air, hot water for dryer equipment or chilled water for cooling process [1].

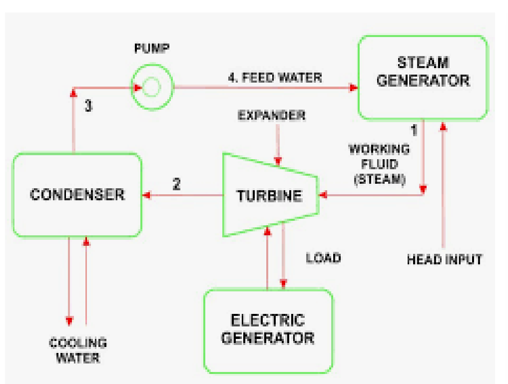

Fig 1.2 Basic cogeneration system

1.3 GAS TURBINE COGENERATION SYSTEM SCENARIO

The cogeneration system has contribution towards the sustainable use of energy directly or indirectly in many ways. These systems provide primary energy savings and have high level efficiency decreasing the greenhouse gas emissions and contribute to the environment and nature. This efficiency results in cost savings, reduce grid congestion and avoids distribution losses also [5]. Steam is being employed for various areas petrochemical industry, food processing industry. The co-generation power plant using steam is a feasible and an economical way of steam production. Cogeneration improves efficiency of plant hence lessening the CO2 emissions. Approximately 10% of the electric power is supplied by cogeneration in the United States [6]. Also combined heat and power systems produces about one twelfth of electricity generated globally. Cogeneration is most widely practiced in eastern and western region of Europe. Denmark being the leader worldwide meets 52 % of its electricity demands (5,690 MW) as recorded in 2003. About 13 % of electricity in Germany (21,203 MW) was produced from cogeneration in 2005. CHP accounts for a significant 19 % of its power by installing CHP capacity of nearly 35,000 MW. CHP can minimize global greenhouse emissions as low as to 4 % in 2015 and 10 % in 2030 as per the reports of the International Energy Efficiency [7].

An important parameter in evaluating any cogeneration system has been power heat ratio and gas turbine based cogeneration system has been proven to provide better performance than any other system. Gas turbine based cogeneration system comes with flexibility of rejecting heat at different temperature which is required for operation of different coupled system and help in increasing the utilization efficiency of the fuel used up to 75% [8]. Gas turbine cogeneration system also provide advantage in terms of overall reduction in CO_2 emission as natural gas powered gas turbine shows a 20% decrease in the overall CO_2 emission as compared to other fossil fuels and 40% decrease in CO_2 emission in compared to coal based power plants [9]. World Energy outlook has predicted an increase in the natural gas consumption in upcoming years in power producing industries as well as in other chemical industries as a source of fuel, it is expected that the natural gas will contribute to around 40% of the total power generation by 2040 [10]. This will further enhance the potential of integration of gas turbine based cogeneration systems into industries. Still, 60% per cent of the energy is likely to be supplied by conventional sources till 2040.

1.4 TYPES OF COGENERATION POWER PLANTS:

Cogeneration power plants are basically grouped based on the order of energy use, and on the basis of technology employed. Classification of cogeneration systems is shown below in Figure 1.3.

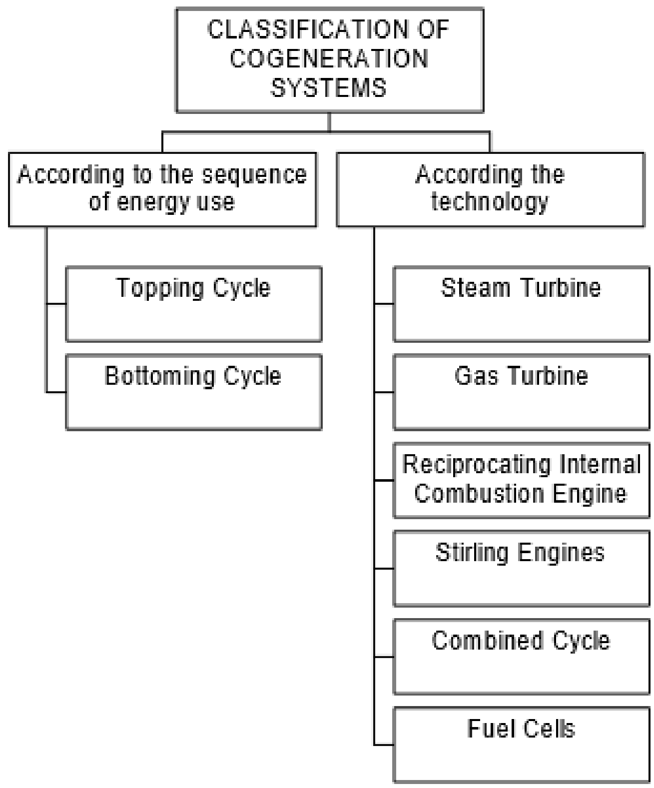

Fig 1.3 Classification of Cogeneration system [11]

1.4.1 Based on series of energy employed:

The topping cycle has the prime mover which is used to produce electricity and excess heat or by-product i.e. steam is utilized in the thermal processes and also may be heating and cooling of an area. Topping cycles are known to be utilizing the fuel efficiently as compared to basic systems used for production of electricity and heat in separate cycle [12].

Topping cycle mainly used for producing electricity with the help of steam turbine. Example is district heating or water desalination.

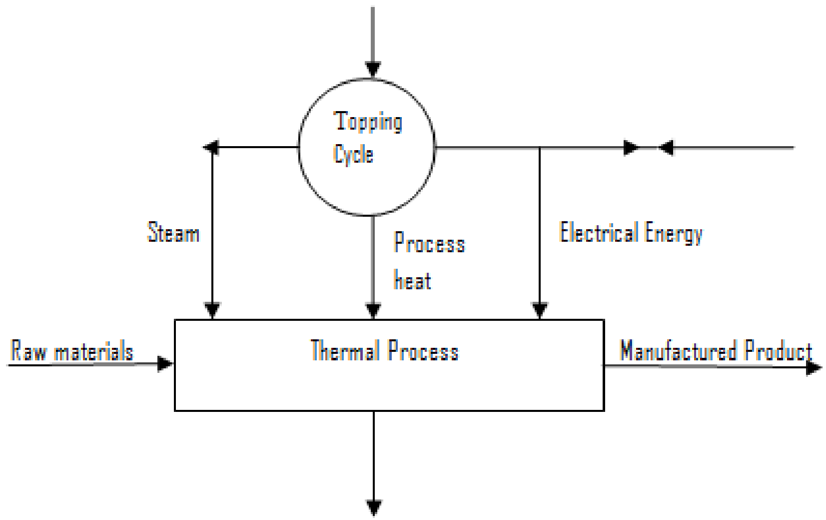

Fig.1.4 Topping Cycle [12]

Bottoming cycle plants are mainly used for producing high temperature heat for industrial processes. This cycle is generally used in industrial processes requiring very high temperatures realized in component such as furnaces used in the glass and metal deployment sectors. Bottoming cycle heat having high temperature that is produced for a process (for e.g. in a furnace of a steel mill or glass-works or a cement kiln) are used either directly for operating a gas turbine generator if in case the pressure is appropriate, or can be used secondarily in a heat recovery steam generator to produce steam that pilots steam-turbine generator. Bottoming cycles are also fuel efficient in contrast to basic conventional plants [12].

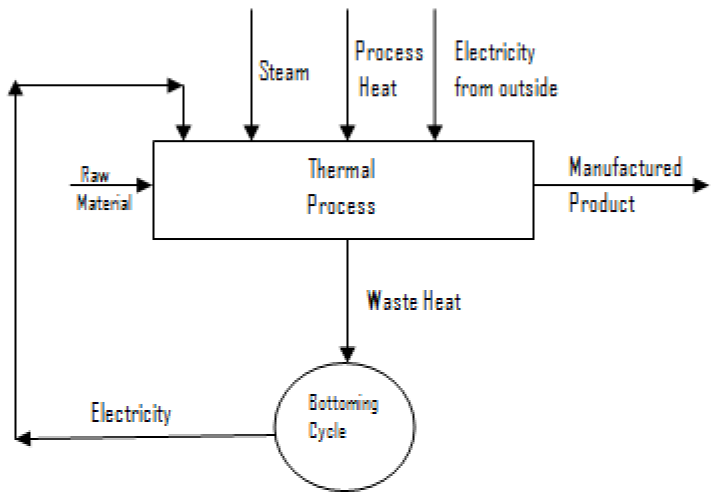

Fig 1.5 Bottoming Cycle

1.4.2 Based on technology:

However there are a number of configurations of the cogenerations systems, there are some configurations which are mostly used and have been accepted. Some of these cogeneration systems are discussed below.

1.4.2.1 Steam turbine Cogeneration system:

Steam turbine cogeneration system has mainly a steam generator, a steam turbine and a thermal reservoir for dumping heat. The cycle in this system is either the basic rankine cycle or with improvised form say with reheat or regeneration. Steam generator also known as boiler is employed with combustion of fuel to generate heat for the production of superheated steam. Nuclear reactors are the alternative for the same purpose.Also the renewable forms of energy such as solar energy or wind energy can be utilized for generation of heat. The working ambient conditions have a larger coverage. Steam pressure when taken into consideration may vary from minimal bars upto 100 bars in cogeneration systems. Higher pressures are observed in the utility sectors. The variation of the temperature of steam has a wide range starting from few degrees to almost upto 450°C. Taking in light the utility sector it may range upto 540°C. Range of the power output is 0.2-100 MW however larger amount can also be produced [13].

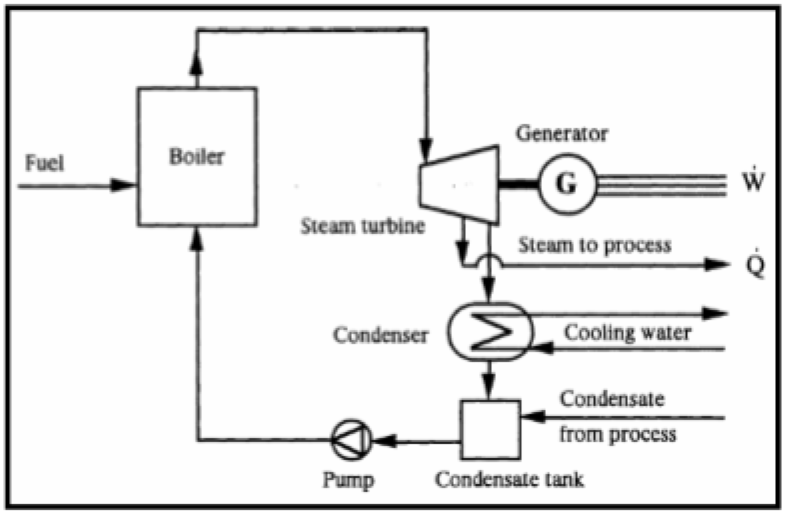

Fig. 1.6 Condensing type steam turbine [13]

1.4.2.2 Gas turbine cogeneration system:

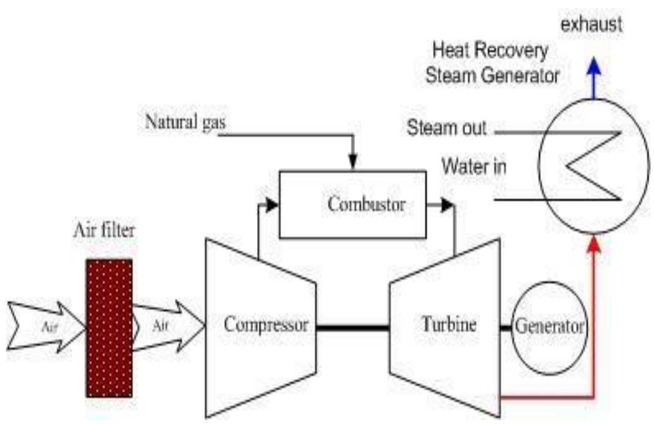

Fig.1.7 Simple gas turbine cogeneration system [3]

Cogeneration with gas turbine is preferred because a simple gas turbine cycle has relatively lower investment cost, more flexible and quite simple in its working with much reliability [3]. The waste energy is reused by combined heat and power system leading to efficient utilization of the resources to prevent any environmental degradation and to attain the targets of better profitable competition. Also air-conditioning district heating can utilize cogeneration also has a dormant in various industries like ethylene production, dairy, refinery or paper, improvised oil recovery systems and chemical industries comprising chlorine producing industries, caustic soda plants and methanol plants based on the natural gas [4].

1.4.2.3 Reciprocating internal combustion engines cogeneration systems

Reciprocating internal combustion engines are highly efficient including the smaller reciprocating engines. They are the most likely opted for the combined heat and power systems for the applications involving the commercial sector or any institutional body or residential area and moreover industrial sectors where a wide range of voltage is essential from low to medium.

The reciprocating engines are classified on the basis of internal combustion engine cycle which may be an Otto cycle or Diesel cycle or dual cycle. In an Otto cycle, a mixture of air and fuel is mixed in the carburetor which is compressed in the cylinder which is combusted by spark plug causing the ignition and thus producing the power stroke. On the other side only air is admitted to the cylinder and compressed air is sparked with the fuel that is injected by fuel injector leading to the spontaneous ignition of the high temperature compressed air [12].

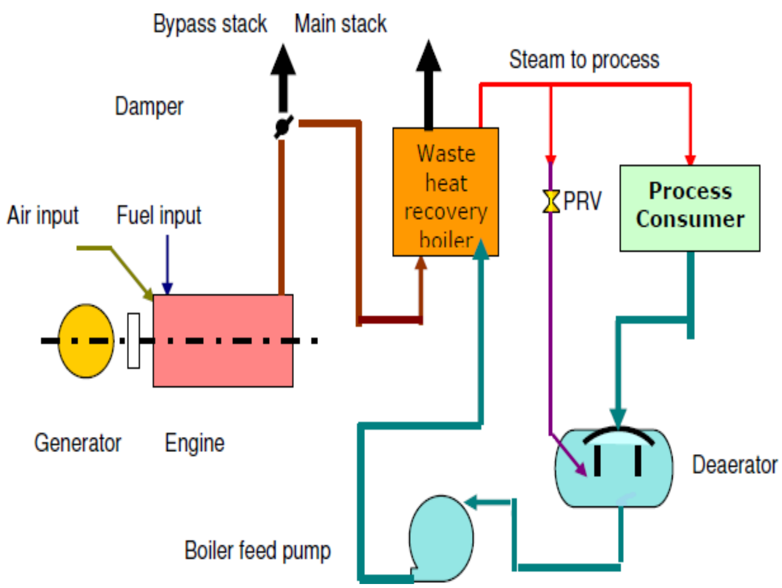

Fig 1.8 Reciprocation based cogeneration system [12]

1.4.2.4 Cogeneration in Stirling engines

These systems are not widely used as these are not well developed yet and need certain improvisation for being fully developed. However they are topic of interest as there are a number of advantages such as scope of great performance at partial loads, high efficiency, lower noise caused by vibration , emissions are low, flexible with the use of fuel.

1.4.2.5 Combined cycle cogeneration systems

The combined cycle plant is composed of two different thermodynamic cycles and these cycles run on a same working fluid and pilots at different levels of temperature. The topping cycle which is a high temperature cycle and acts as a thermal source which is rejecting heat to the low temperature cycle known as the bottoming cycle after being recovered to generate an additional mechanical energy or electrical power. Hence, the efficiency of such cycle increases.

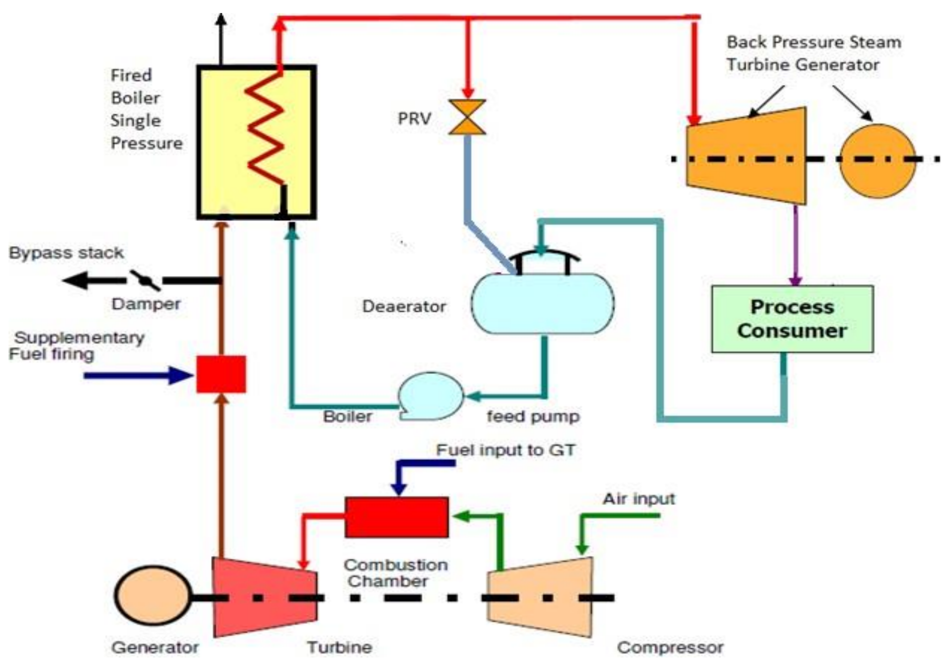

Fig.1.9 Combined cycle cogeneration system

The two most widely used configurations of combined cycle cogeneration systems are as follows:

1. Combined Brayton–Rankine cycle systems

Generally used combined cycle is one which is a combination of gas turbine i.e. Brayton cycle and steam turbine (Rankine cycle) as shown in figure. The accumulation of power i.e. power per unit volume of the combined cycle is more than that of the simple gas turbine as well as steam turbine cycle .The turbine used for generation of steam is a backpressure turbine but we can use condensing turbine as well.

2. Combined Diesel-Rankine cycle systems

Combination of the rankine cycle and diesel cycle is also possibility. Rankine Cycle improves the efficiency of the medium to higher power and makes it economically liable. It is quite similar to the combined cycle plant. The only difference is that there is diesel engine in place of gas turbine system.

1.5 APPLICATION OF COGENERATION SYSTEM:

Cogeneration in industrial applications can be seen particularly in sugar industry, textile mill, food-processing companies, pharmaceutical industry, steel and glass industry and oil refineries that requires both heat and electricity in considerable quantity.

Institutional, commercial and industrial applications are to be established in smaller systems on the basis of packed units. These cogeneration systems are found in restaurants, leisure centre, hospitals, residential complexes and offices. The airports, commercial buildings, offices or large housing complexes use district heating facilities. One of the feature of a heating system is that it drives district heating in the industries and acts as option to use a various fuels for suiting economical, environmental or strategic priorities. The heat provided by cogeneration is ideal for space heating and for providing hot water for domestic, commercial, or industrial use. Large scale applications are classified on the basis of a technology that is same as the cogeneration systems employed in gas turbines, industry or reciprocating engines which are large in size. These systems have a scope in large scale hospitals, huge office complexes, college or universities.

CHAPTER 2

LITERATURE REVIEW ON GAS TURBINE COGENERATION SYSTEM

2.1 INTRODUCTION

Gas turbine cogeneration system is the preferably used system as it has the characteristics such as highly flexible, high reliable and quicker operation as in fast loading and quick start, cheaper and simple in use also. Adding to its advantages is its compactness requiring lesser number of workers and is voluntarily available. Because of the several perks of the simple cycle gas turbine, there is compromise with the efficiency restricted at part loading conditions and also superiority of the fuel cost on generation The gas turbine is known for its curing effects on environment exhibited in lessening the air pollution which results in curbing t a very severe problem of global warming [16]. Efficiency of the gas turbine can be improved possibly by increment in the turbine inlet temperature limited by the metallurgical consequences of the turbine blades. The utilization of the heat contained in the exhaust gases is done in cogeneration system for production of electricity and heat. The heat energy in the exhaust gases of gas turbine can be utilized for drying purpose, as a process heat, steam generation in boilers and to preheat the combustion air. The prospective cogeneration utilization industries may include petrochemical, textile, paper, board and agriculture sector and also metal industries and the air condition and district heating. Cogeneration can be used in number of probable activities. It includes expanding the established facilities, developing new facilities in the industries replacing the outdated steam production equipments. The only purpose of gas turbine cycle is the using the exhaust to transfer heat to the working fluid and further to the electricity which is is the main production as it is valuable. The other part left out is rejected in the atmosphere as heat. It can also be observed some industries utilize process heat as the input form of energy. The process heat can be manipulated in heating spaces, air conditioners, desalinators and absorption chillers. It is an essential requirement to counterpart the exhaust heat temperature associated with application due to the losses in exergy and irreversibility. Major objective in the coming years is to enhance the performance of the several configurations of gas turbine cycles. The purpose to achieve a target of the industrial gas turbines with minimal 50%

efficiency, combined cycle efficiency above 60%, small scale gas turbine system efficiency atleast 35% and optimize the designs for employing fuels less than 25% of the heating value as in natural gas. The combined heat and power plants have utilization factors reaching a high mark of 85-90% i.e. larger systems providing 40% of electrical power and 50% of heat energy whereas smaller systems provide 30% electrical and 60% heat energy opportunities [14].

2.1.1 CONFIGURATION OF GAS TURBINE COGENERATION SYSTEM

The gas turbine has Brayton or joule cycle associated with it. In the thermodynamic cycle the process starts with atmospheric air being compressed, introducing the fuel and then proceeding with combustion followed by the expanding of the high temperature gases in the gas turbines as well as power producing turbines. The power produced by the change in enthalpy in turbine is utilized for piloting the compressor and also the power generator. The two types of gas turbines according to the system design are classified as aero derivative and the industrial gas turbines.

Aero derivative gas turbines for consistent power are inspired from the jet engine dynamics. These turbines are thermally efficient and light in weight but capacity is quite low and a limitation. The highest aero derivatives are about 40 MW in capacity, simple cycle efficiencies are about 45% and the compression ratios employed are approximately 30:1. Also, Industrial gas turbines are more robust in comparison having capacity of range from 1 MW to 250 MW. They can operate for longer time during the rework period, and are not appropriate for intermittent load operation but for continuously operating base load. The drawback is that they are bulkier and has lesser efficiency compared to that of aeroderivatives. The compression ratios are as high as 16:1 and do not need an external compressor so frequently. The efficiency being approximately about 40%[12].

Configurations of Gas Turbine Cogeneration Cycle:

- Open Cycle Gas Turbine Cogeneration Cycle
- Closed Cycle Gas Turbine Cogeneration Cycle

2.1.1.1 Open cycle gas turbine cogeneration system

The open cycle gas turbine is air is taken from the atmosphere to the compressor and its temperature and pressure is increased and admitted to the combustor. The compressed air enters through a diffuser into a constant-pressure combustion chamber. In combustion chamber fuel is burned for transferring heat to the air. The diffuser is for reducing the air velocity to acceptable values in the combustor. Combustion takes place in the presence of excess air. This point gives the highest temperature of the cycle. Greater the temperature, greater is cycle efficiency. The high temperature and pressure exhaust gases will enter the gas turbine. The production of mechanical work in the gas turbine takes place for driving the compressor as well as the load.

The exhaust gases leaving the turbine possess a considerable amount of temperature (450-600°C), which is ideal for the high-temperature heat regenerator. The steam produced has high quality making it suitable for driving a steam turbine using additional power also adding heat for thermal processes [12].

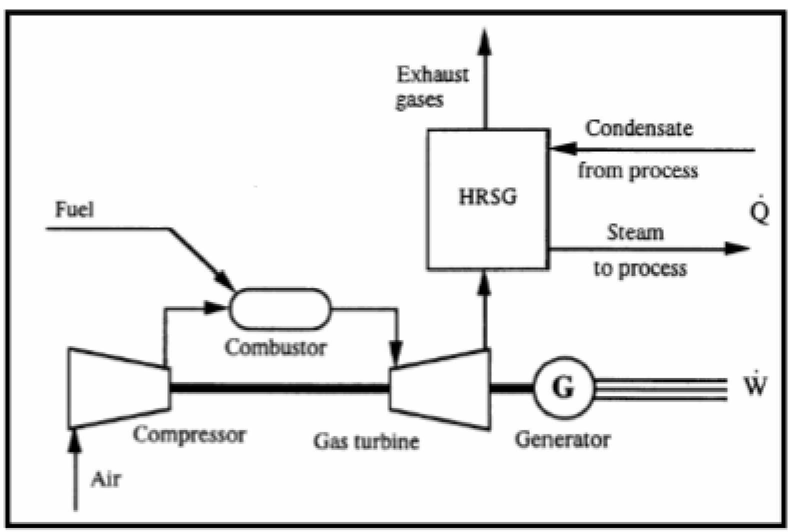

Fig.2.1 Open cycle gas turbine cogeneration [12]

2.1.1.2 Closed type gas turbine cogeneration cycle

The working fluid which may be air or helium circulates in a closed cycle. The working fluid gets heated in the heat exchanger and then enters the turbine where the the fluid is cooled down releasing required useful heat. The working fluid thus remains pure and there is no erosion. The source of heat may be the external combustion of the fuel or the waste from industries or other forms of energy i.e. solar energy or nuclear energy can be used. After study it can be concluded that closed cycle cogeneration systems is assumed to be as reliable as that of open-cycle cogeneration system. However, the availability is assumed to be higher because of the use of the clean working fluid.

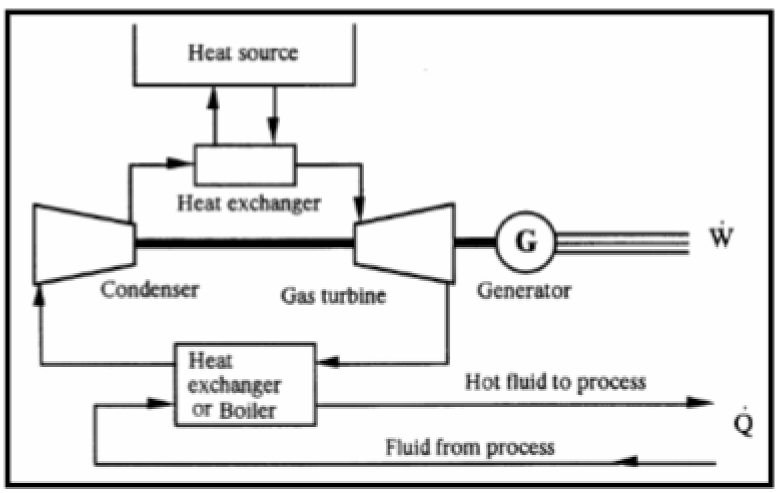

Fig.2.2 Closed cycle gas turbine cogeneration system [12]

The selection of the gas turbine between the open cycle gas turbine systems and the closed cycle gas turbine system is based on comparison presented in Table 2.1. It presents a comparison of the advantages and disadvantages of the simple cycle (without regeneration or additional improvements) of both open cycle and close cycle systems.

Table 2.1 Comparison of Open and Closed Cycle Cycle [15]

Types of System	Open Cycle Gas Turbine	Closed Cycle Gas Turbine
Advantages	Most available gas turbine system in any sector of operation in current scenario. It is simple in design. Temperature of heat addition is higher. Also, thermal efficiency is higher.	The working fluid is clean. No problem of erosion and corrosion. Higher efficiency in part-load operation
Type of system	Open Cycle Gas Turbine	Closed Cycle Gas Turbine
Disadvantages	Thermal efficiency reduces when the load is drops down below the design point. Contaminant emissions are considerable in this case.	Investment and maintenance cost is very high. Components should be structurally suitable for high temperatures and pressures of the system. Special management system for working fluid. Temperatures are considerably high in the heat exchanger. Temperature at which maximum heat allowed to be added of a closed cycle gas turbine as compared to that of open cycle gas turbine is considerably lower. Thus, due to this reason limitation exists in maximum power conversion efficiency.

From the above comparison a simple open cycle gas turbine cogeneration system takes over and appears as a better option, as it gives the simply designed technology with the highest thermal efficiency and the possibly lowest cost of capital as well as costs of maintenance [15].

2.2 LITERATURE REVIEW

2.2.1 REHEAT AND REGENERATIVE CYCLES

M.S. Zaamout, Y.S.H. Najjar [17] in their study gave emphasis on the fact that the gas-turbine engine has somewhat poor performance at part-load conditions. The power output degrades during the hot weather conditions. Thus, regenerative cycles utilizes the exhaust waste energy in the recuperator with the use of water injection. As a result, the water-heater is assumed to recover the reduced power and enhance the fuel economy considerably lower cost and with higher reliability in comparison to that of the combined cycle. The performance of the evaporative regenerative cycle (ERC) is studied with the consideration of parameters in this research work taking the compressor pressure-ratio Rc, turbine inlet temperature T_{03}, and water air ratio w as the main variables. The analysis in the research work was carried out with the help of a specially tailored designed computer program where main variables are varied in wide ranges. The result showed in the figure 2.3 and 2.4 shows that ERC exceeds the performance of RC by about 57% approximately in power and 13% in efficiency.

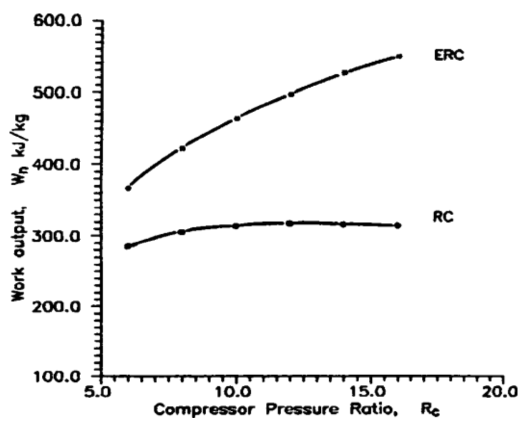

Fig.2.3 Variation of work output with R_c [17]

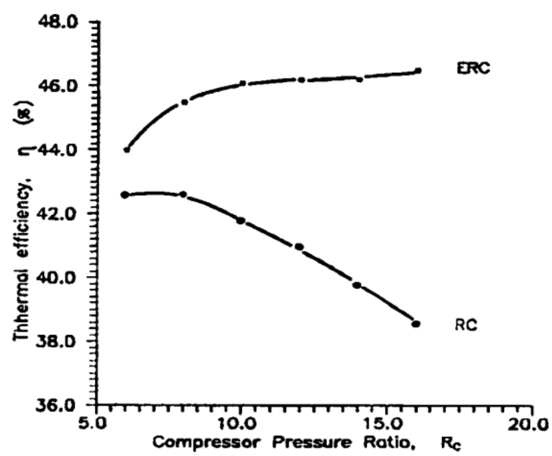

Fig.2.4 Variation of thermal efficiency with $R_{c[}$ [17]

Y.S.H. Najjar [18] evaluated the low-pressure steam-injected intercooled gas turbine with cogeneration system. Combined power plants are expected to be the ideal solution to cope up with the increasing demand for electrical power as well as the heat. These plants provide unique flexibility over wide ranges of loading. The ability to use a variety of fuels of the gas turbine may become superior to all expanding electricity providing systems. In this research paper work a system of turbo-STIG which is intercooled low-pressure steam-injected gas turbine with cogeneration is compared with an intercooled cogeneration system.

A.Khaliq, S.C. Kaushik [19] presented thermodynamic method for evaluating the performance of reheat combustion gas turbine cogeneration system. They studied the process steam pressure and pinch point temperature effects on exergy and energy efficiency to be utilized in the design of reheater between turbines and HRSG. The results which are obtained in the study bring out the conclusion that the power to heat ratio increases with enhancement in pinch point. However, the first-law efficiency and second-law efficiency degrades as the pinch point increases. In case of power to heat ratio and second-law efficiency increases considerably with the increase of process steam pressure whereas the first-law efficiency decreases with the process steam pressure.

Thus, the reheat in the cycle provides considerable enhancement in the process heat, electrical output, production, fuel-utilization efficiency (first law efficiency) and second-law or exergetic efficiency. The schematic diagram shows the energy flow of reheat cogeneration system in fig.7.The parametric variation of power to heat ratio, fuel utilization efficiency with respect to the pinch point temperature and process steam pressure is given below in fig.2.5 [19].

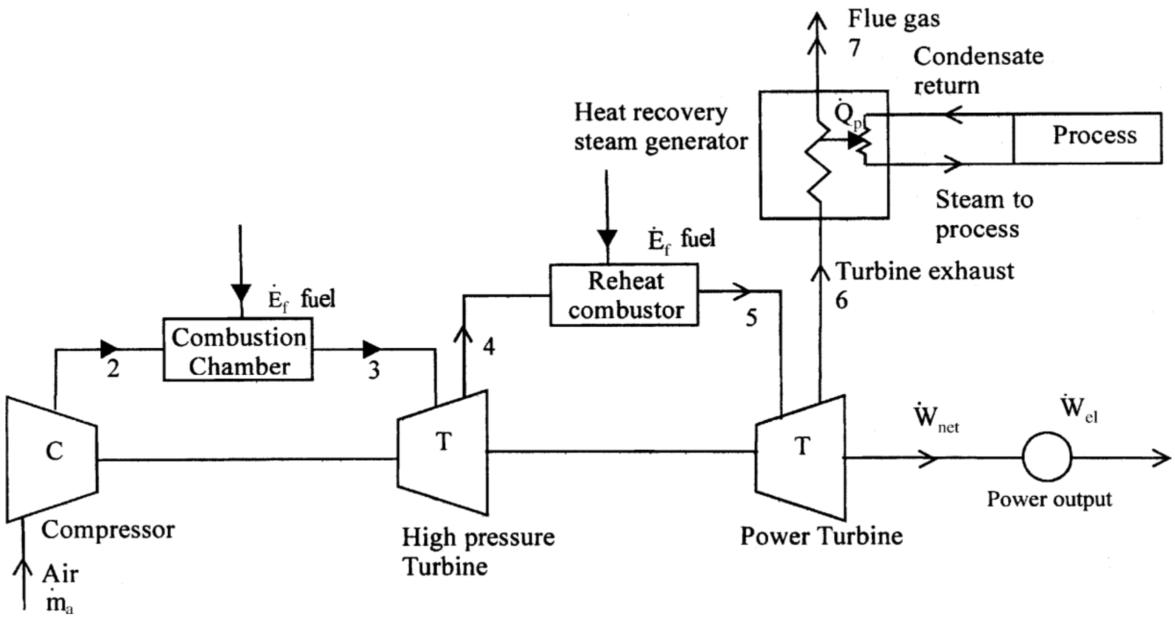

Fig.2.5 Reheat combustion gas turbine cogeneration system [19]

An investigation is presented in the research work by A.Khaliq, K.Choudhary [15] which studied the combined first law as well as second law analysis for cogeneration system employing the inlet air cooling with evaporative aftercooling of discharge from compressor.Thus, Computational analysis is provided to analyze and study the effects of the overall pressure ratio r_p, and relative humidity effecting the exergy destruction in every component, turbine inlet temperature (TIT), power-to-heat ratio (RPH), first-law efficiency and also second-law efficiency of the cycle. Thermodynamic analysis states that exergy destruction in every component of the cogeneration system is considerably affected by turbine inlet temperature and overall pressure ratio and ambient relative humidity has merely any effects.

It is noted from the study of research work the exergy destruction in every component of the cogeneration cycle is impacted by the overall pressure ratio whereas the turbine inlet temperature and the ambient relative humidity have almost no effect. During the combustion process, the maximum exergy is destroyed which shows that approximately 60% of the total exergy destruction in the whole system. The second exergy destruction on a larger scale happens in the compressor, regenerator, evaporative after cooler, intercooler, and also in inlet air cooler [20].

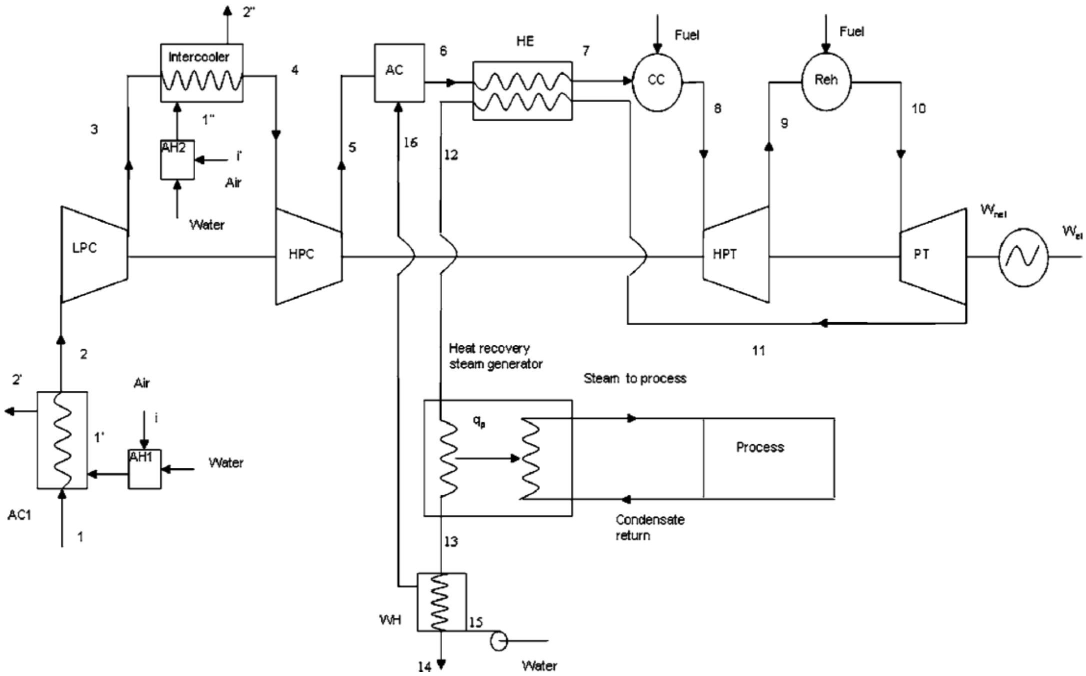

Fig 2.6 Schematic Diagram of the gas turbine cogeneration system with inlet air cooling and evaporative aftercooling of the compressor discharge [20]

A series of small reheats concept in a gas turbine system is introduced by Norihiko Iki. The gas turbine reheat consists of Brayton cycle, inverted Brayton cycle and heat exchangers. The system is supplied with hot water and steam. The components present in the system can be interchanged with the micro gas turbine. Main focus in the research work was the small reheat gas turbine system performance keeping in view it as a cogeneration system. The efficiency of the electrical power output is significant over 31% and the heat output efficiency is only 16–21%. So, efficiency is about 51% overall. Thus, small reheat gas turbine system heat recovery is enhanced. Thus, it can be concluded that the overall efficiency is over 60% and cooling water is consumed less and reduces significantly [21].

Trigeneration is the production of the heating, cooling and turbine work output simultaneously from one system. It accentuates the economical benefits as well as the benefits to the environment of combined heat and power generation by the heat produced from the Cogeneration unit to act as driving power for the cooling unit. J.Godefroy [22] presented his work on based on gas engine mini Combined Heat and Power system (5.6 KW) based on the designing and analysing of the trigeneration system i.e CHP unit with an ejector cooling unit. The analysis showed around 50%

of overall efficiency can be gained. For the system requiring both heating and cooling the above efficiency is obtained. However, CHP electrical output on one hand enhances the cooling capacity of cooling cycle. On the other hand this does not boosts the overall efficiency and also plays an important role in increasing the CO_2 emissions. Emissions savings compared to traditional systems could be achieved with improvements of the heat transfer from CHP to cooling cycle [22].

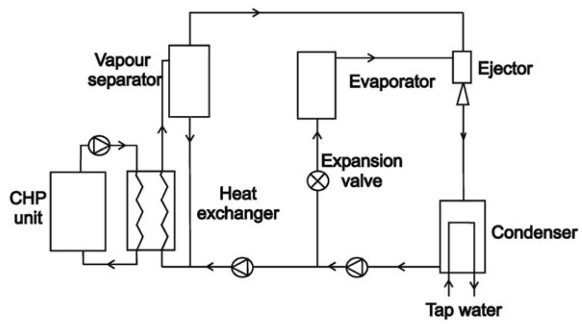

Fig 2.7 CHP-ejector system [22]

Overall efficiency of simple combined heat and power system is as low as 28% to 30% but there is significant overall efficiency increment that is nearly about 19 % when system is designed for heating as well as cooling purpose simultaneously. Thus, designing basic trigeneration systems to fulfill the need for producing heating and cooling simultaneously would be thus preferred [22]. Thus, making use of part of the electrical output to enhance the temperature of the generator improves the cooling capacity by 60% for the basic cooling system as compared to the cooling system using a part of the heat for the heating purpose as well, and it boosts upto 99% with trigeneration system in comparison to the basic cooling unit. This is correlated to an improved generator temperature which converts into corresponding enhanced coefficient of performance of 50% and 75%. A loss in the net electrical output however compensates for the increased COP .Therefore, the effect on the trigeneration efficiency is nullified as a whole [22].

The first basic system makes use of all the available heat from one combined heat and power unit for the cooling system unit. Here, it is assumed that a generator temperature of 70 °C can be achieved. Second, third and fourth system is the representation of variations in the basic system in which some part of the CHP output is used for heating ,and another part of the electrical power output of CHP unit is utilized to enhance the generator temperature.

TABLE 2.2 COMPARISON OF FOUR VARIANTS OF TRIGENERATIO SYSTEMS[22]

Systems	**1**	**2**	**3**	**4**
Gas input (GCV) (kW)	22.9	22.9	22.9	22.8
Heat output (kW)	0	5.01	0	5.01
Additional heat input to the cooling cycle (kW)	0	0	1	1
Electrical output (net) (kW)	5.3	5.3	4.3	4.3
Generator temperature (°C)	70.1	70.1	84.4	93.6
Ejector cooling COP	0.145	0.135	0.202	0.226
Cooling output (kW)	1.58	1.01	2.62	2.012
Electrical efficiency (%)	22.4	22.4	19.0	19.0

4.2.2 INLET AIR COOLING SYSTEMS

It is fact that the gas turbine efficiency is comparatively lower at the design conditions .Also when there is increment in ambient air temperature; the efficiency degrades more at part load conditions as well as off-design condition. Nowadays, improvement in power output and efficiency can be expected by enhancing turbine inlet temperature and engine pressure ratio However, this result in increment of NOx as it is undesirable pollutant. Therefore, interestingly cogeneration has become a popular term in gas turbine plants. Approximately, 30% of the gas turbines in stationary state are installed nowadays is cogeneration applications. Earlier, nearly 5 years back, out of the machines which were deployed in cogeneration applications, 10% of them comprises of addition of an inlet air-precooler which is extended to the evaporator of an aqua ammonia absorption chiller further being driven by the last heat retrieved from the gases exhausted by the engine. A heat recovery boiler is basically employed to recover a part of the exhaust heat before its entrance into the the generator of the chiller. The performance of the system proposed in the research work is analyzed in terms of the power, specific fuel consumption and efficiency and it is compared with the simple basic cycle. Variables in the parametric analysis of the system are compressor pressure ratio, ambient temperature and turbine inlet temperature. Results obtained reveals that that the proposed combined system has gained power as well as efficiency and specific fuel consumption of about 21, 37 and 27% respectively [23].

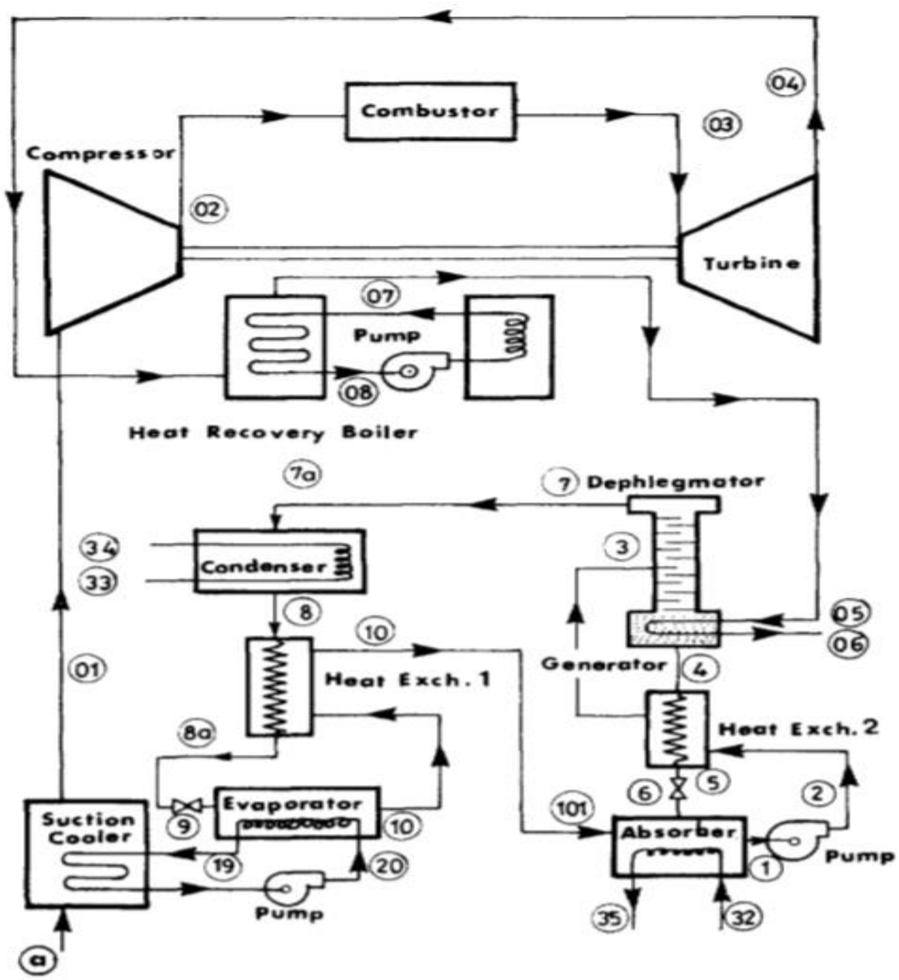

Fig2.8 Schematic representation of the combined system with precooling [23]

The dimensional parameters and operating characteristics of the optimally designed Maisotsenko indirect evaporative cooler for the purpose of pre-cooling the inlet air to the compressor and as a result generation of power is enhanced, also results in the best strategy for investment. Net power mix is used to create four optimization techniques for power generation system used in gas turbine. In this work, enthalpy difference and payback period time of exhaust gases as compared to the every gas-turbine's reference system, the power generation system being the objective functions. Taking all the cases separately each system is optimized with the different approximate percentages of the allocated investments to the development and research of the project. The implication of plan was in Montazer-Ghaem combined cycle power plant in Iran. The results revealed that the average of the enthalpy difference and net power of the exhaust gases anually should be hiked as compared to the reference state were the objective functions This optimization project had the payback period time of 2.7 years approximately and it also enhanced the annual average of gas-turbine's net power generation system and enthalpy difference of the exhaust gases compared to the reference state 6.02 and 8.92% respectively [24].

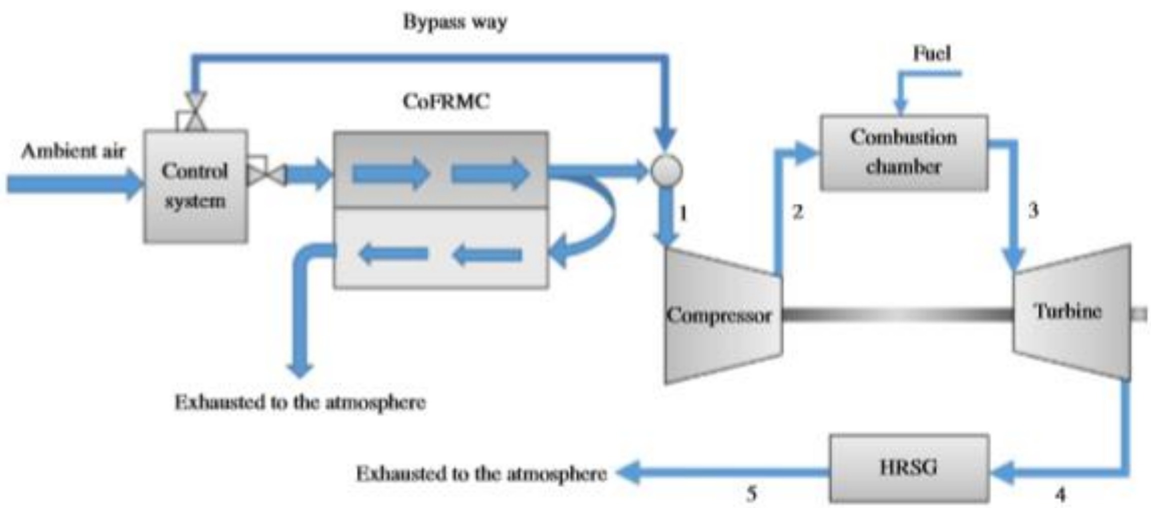

Fig .2.9 Improved GTPGS with the CoFRMC and the suggested control system [24]

A gas turbine inlet air cooling technique is a great option for increasing power output in case for meeting the significant power demands causing highest electricity prices in certain geographical regions during the warmer months. Inlet air cooling enhances the power output which allows higher mass flow rate in gas turbine due to decrement in the compressor inlet temperature.

Industrial gas turbines operating at constant speed can be defined as the combustion machines with constant volume flow rate. As it is fact that the specific volume of air is directly proportional to temperature, thus at constant volumetric rate the increase of the air density results in grater mass flow rate. As a result, the power output increases.

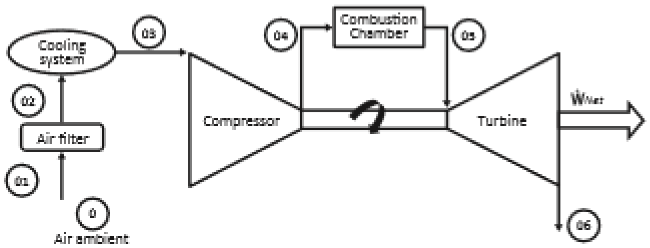

Fig.2.10 Schematic representation of the gas turbine with cooling system [25]

The various methods which are available there for degrading the intake air temperature of compressor. Currently there are basically two systems at hand for inlet air cooling. The foremost and highly cost operative system is the evaporative cooling. The evaporative cooling system uses of water evaporation to lessen the inlet air temperature of gas turbine. Another system uses two ways for cooling the inlet air i.e. Absorption and mechanical compression. The method uses the cooling medium; a heat exchanger is located in the inlet duct for removing the heat from the inlet air. Santos Claudia carried out a thermodynamic analysis to study the gas turbine performance for calculating the heat rate, power output and thermal efficiency .These performance parameters are calculated at various intake temperatures of air and ambient relative humidity. These results are compared with the basic case. The result showed that among the absorption and chiller systems, the absorption chiller has the most annual energy generation with cost of energy being lower. Also, evaporative cooler was found to offer the lowest energy cost however, limiting cooling potential [25].

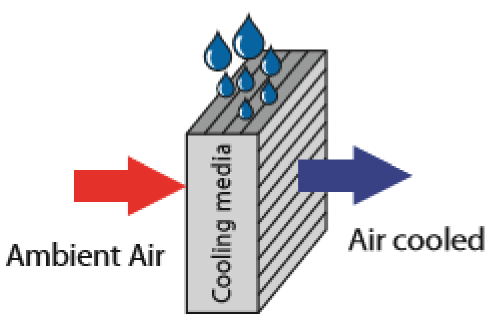

Fig.2.11 Schematic diagram of evaporative coil [25]

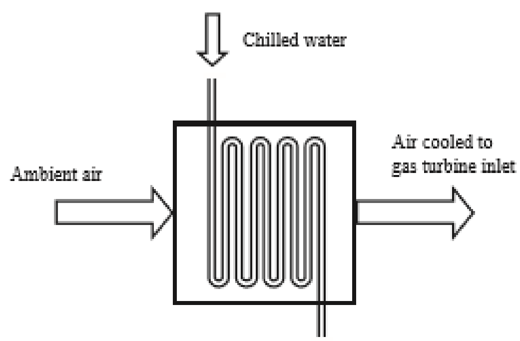

Fig.2.12 schematic diagram of chiller coil [25]

Production of combined cooling, heating, power and water aka CCHPW in an integrated plant was taken under investigation by Sepehr Sanayea. A gas turbine was included in the study with steam injection in the combustion chamber, a Heat Regenerating Steam Generator (HRSG) for production of steam and absorption refrigeration system used for space as well as inlet air cooling. The Energetic, Exergetic, Economic and Environmental model designing and optimizing the consolidated CCHPW plant was experimented for studying design parameters. The optimization of is done by the genetic algorithm method to calculate exergetic efficiency as well as the total annual cost. Temperature of inlet air to the compressor was 18.6°C and injection of steam into combustor employed in gas turbine was around 1.81%. The second law efficiency and Total Annual Cost (TAC) of consolidated CCHPW plant at the optimum point was 30.7% and 28.5 million dollars approximately, respectively and the payback period about 4.38 years was estimated for the plant which was analyzed at the optimum conditions [26].

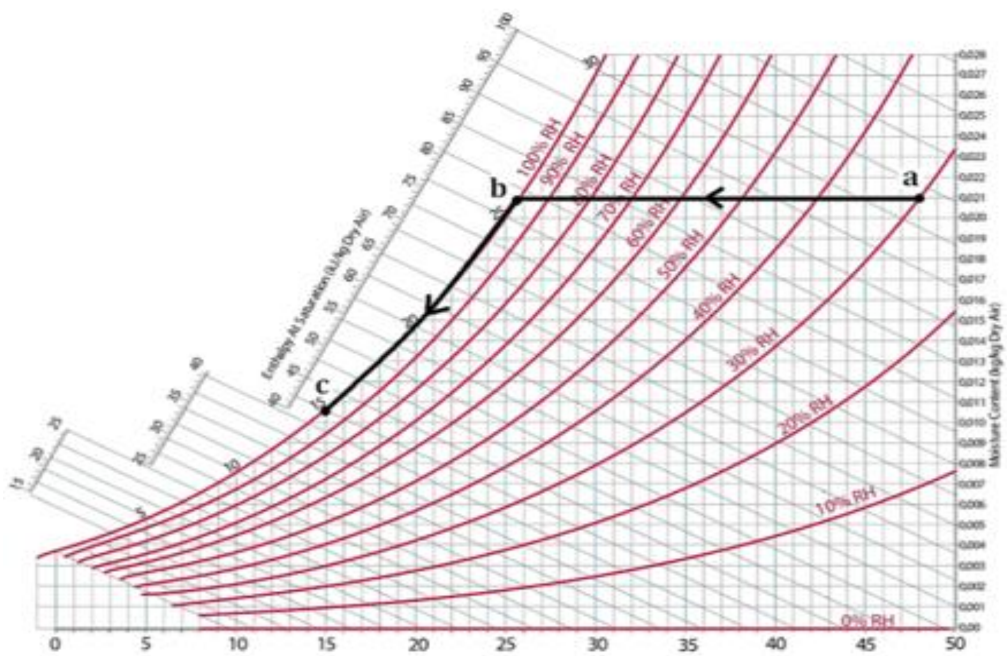

Fig.2.13 Compressor Inlet Air Cooling process in psychometric diagram [26]

CHAPTER 3
THERMODYNAMIC MODELLING OF COGENERATION SYSTEM WITH REHEAT AND INLET EVAPORATIVE COOLING

3.1 INRODUCTION

In thermodynamic analysis of gas turbine cogeneration system with inlet evaporative cooling and reheating, first the thermodynamic model of the two system i.e., first is the reheat cogeneration system and second the gas turbine cogeneration system was prepared separately and individual models were validated from the work carried out in previous paper. The cogeneration system with reheat was validated separately from A.Khaliq (2004) [19] and gas turbine cogeneration system with inlet evaporative cooling was validated from Ali Sohani, Yashar Farasati (2017) [25]. Then the two systems were integrated to perform the parametric study of gas turbine cogeneration system with inlet evaporative cooling and reheating.

3.2 SYSTEM DESCRIPTION

Fig 3.1 shows gas turbine cogeneration cycle with reheat. The components and the system operating on this cycle as a whole is shown in the figure. In this cycle air at atmospheric pressure enters a compressor at state 1 and it is then compressed from state 1 to state 2 in the compressor. On going through the first combustion chamber fuel is added to enhance the temperature at constant pressure to state 3. The hot gases are then expanded in the first turbine to state 4. Now again fuel is added in a second combustor to reheat the gases again to the state 5. After this the reheated gas is expanded through a power turbine for driving a load. Hot exhaust gases from the second turbine which is a power turbine is the waste heat source for producing the process heat.
It can be observed from the description of the system that the energetic and exergetic parameters of process in the cycle are affected by a number of parameters.

Thus, it is expected that the temperature of hot exhaust gases, temperature of saturated steam in HRSG at greater temperature, pinch point as well as the pressure of process steam will be have a major effect on the performance parameters.

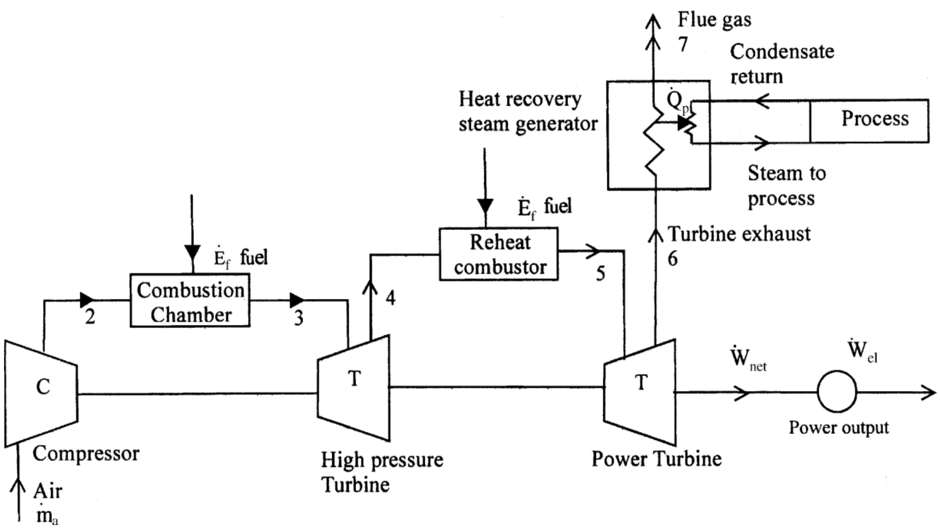

Fig 3.1 Reheat combustion gas turbine cogeneration system [19]

The output of turbine depends strongly on the ambient air temperature with corresponding drop in power output being 0.54-0.9% for every 1˚C increase in the surrounding temperature (0.3-0.5 % per 1˚F). Gas turbines can however tend to realize drop in power output nearly about 14-20% when increment in ambient temperatures is from something around 15˚C (59˚F) to 35˚C (95˚F).

Power output = f (Ambient temperature)

Temperature rise	Drop in power output
1˚C	14-20%

So, there seem to be need for the inlet air cooling before the compression and that can be achieved by inlet evaporative cooling.

Fig 3.2 shows a diagram showing the system which consists of the inlet air cooler along with the gas turbine power plant. The gas power plant further has compressor, combustion chamber and the turbine as its components. Here, in this reference work three inlet air cooling systems were studied namely evaporative cooling, absorption and mechanical chiller and they are compared for gas turbine performance. Each case is also compared with the base case i.e. evaporative cooler was found to be a simpler system and more effective in the hot dry areas as it makes use of latent heat of vaporization to cool the ambient temperature from dry bulb temperature to the wet bulb temperature. The process involved in this cooling method utilizes the conversion of the sensible heat into the latent heat, which is the ambient air cooled by the evaporation of the water from the panel wet surface i.e. the cooling media. The working fluid which is air is cooled from the state 2 to state 3 in inlet air cooling system. The air is compressed to high temperature and pressure from state 3 to state 4. The working fluid after combustion in the combustion chamber is flue gases entering in the turbine. In the turbine, the working fluid expands to produce work from state 5 to state 6 [25].

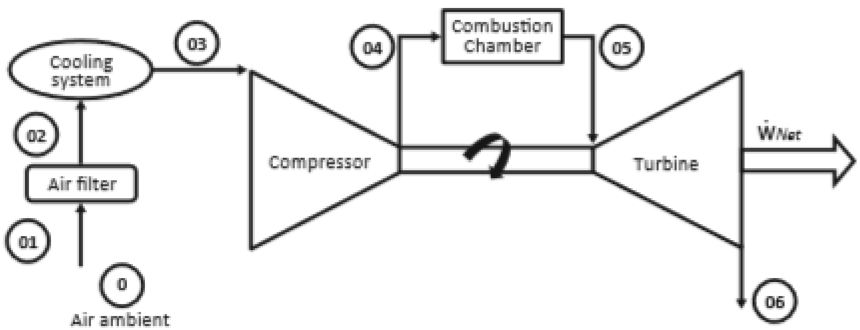

Fig 3.2 Schematic diagram of gas turbine with inlet evaporative cooling system [19]

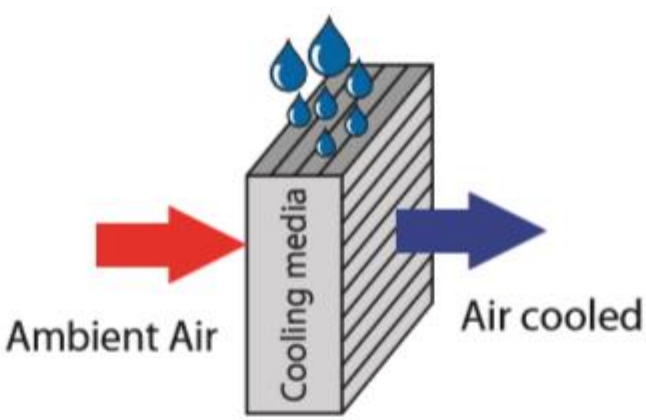

Fig 3.3 Schematic diagram of evaporative cooling medium [19]

Fig3.4 represents a proposed gas turbine cogeneration system with inlet evaporative cooling and reheats. The advantages of the reheat cycle and inlet evaporative cooling are to be analyzed for better efficiency and specific work output. In this system basically some important components and processes are explained for better understanding of the thermodynamic aspects. The main components include air cooling system and gas turbine cogeneration system with reheat. The inlet evaporative cooling system consists of air-filter and evaporative cooler. The air filters are used to filter the ambient air and removes unwanted dust particles and particulates to avoid choking in cooling system from state 1 to state 2. The cooling system is generating the droplets of sizes 5–20 mm that has been injected into the air stream where these evaporate and responsible for air cooling. The data results from test have shown that this process can be as effective as 100% (i.e. wet bulb temperature can be reached) even in harsh -humid regions. Gas after cooling is compressed from the condition 3 to 4 to high temperature and pressure. This air is then passed through the combustor where fuel energy is added and heat is given to the air from 4 to 5.After the addition of the heat energy it is expanded in high pressure turbine 5 to 6. The hot gases is now expanded to state 4 and further fuel addition takes place in the second reheat combustor to get gases heated to the state 7, then after this process the reheat gases expands in a power turbine from state 7 to state 8 for driving the load. The source for process heat production is the waste heat produced by the hot exhaust gases from power turbine.

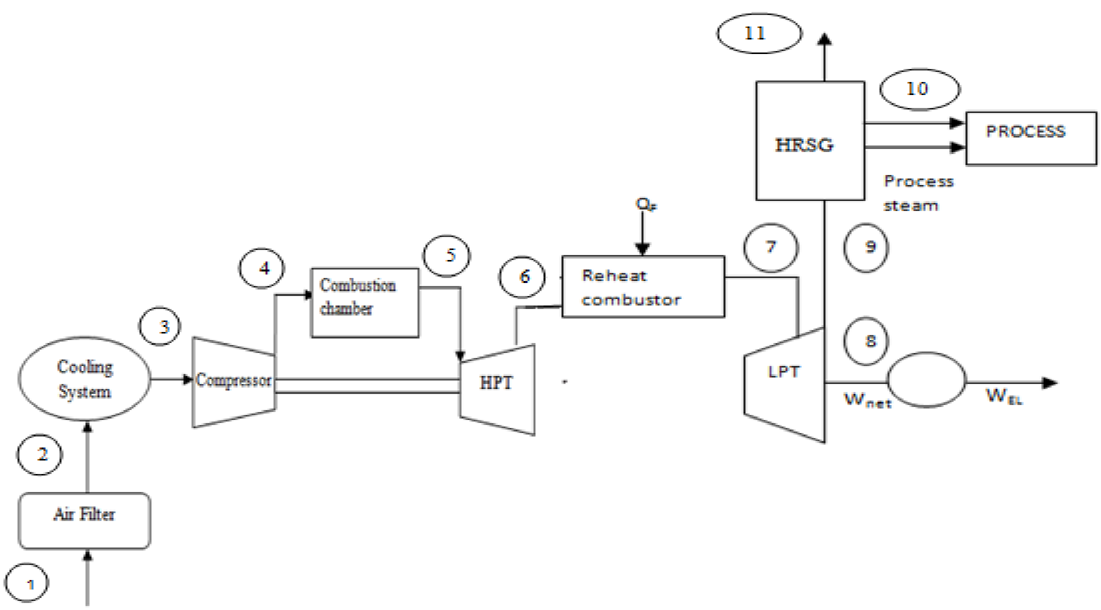

Fig 3.4 Schematic diagram of gas turbine cogeneration system with inlet evaporative cooling and reheat (Proposed work)

3.3 THERMODYNAMIC MODELLING

The following assumptions were considered while modeling and analyzing all the systems i.e. reference and proposed system:

1. All processes in the cycle are assumed to be in steady state.
2. Analysis is carried out considering air compressor and gas turbine as adiabatic.
3. Fuel used is natural gas with Lower heat value is 48.235 MJ/kg.
4. The working fluid air which passes through the compressor is assumed to behave in a similar manner to an ideal gas.
5. Working fluid in the turbine is assumed to be flue gases.
6. The ideal gas mixtures principles are followed for air as well as combustion products.

3.3.1 Thermodynamic modeling of Gas turbine cogeneration system with reheat

Thermodynamic analysis of this cogeneration system is carried out with the help of energy equations. These equations are taken from the reference paper A. Khaliq [2004] and are valid for the cycle in figure 3.1.

Main performance parameters that are calculated are as follows:

1. Fuel Utilization efficiency (η_f): It is defined as all energy in the useful products ($\dot{W}_{el}$ and $\dot{Q}_{el}$) divided by energy of fuel input ($\dot{Q}_f$) .

$$\eta_f = \frac{\dot{W}_{el} + \dot{Q}_{el}}{\dot{Q}_f} \qquad \dots (1)$$

2. Power to heat ratio (R_{PH}): It is the ratio employed to analyse performance thermodynmically and can be numerically written as:

$$R_{PH} = \frac{\dot{W}_{el}}{\dot{Q}_{el}} \qquad \dots (2)$$

In above equations since both the equations are just concerned with quantity but not quality. Both power and process heat is equalized considering quantity only. Therefore, these are called First law efficiencies.

3. Second law efficiency: This gives the quality of energy and tells that electric power has greater value than process heat. It is given as follows:

$$\eta_{II} = \frac{\dot{W}_{el} + \dot{B}_P}{\dot{B}_F} \qquad \ldots (3)$$

Where exergy content of process heat is given by $\dot{B}_p$ and $\dot{B}_F$ is the exergy content of fuel input respectively.

Exergy factor for the process heat is given by:

$$\varepsilon_p = \frac{\dot{B}_P}{\dot{Q}_P} \qquad \ldots (4)$$

Exergy factor for the fuel input is given by:

$$\varepsilon_F = \frac{\dot{B}_F}{\dot{Q}_F} \qquad \ldots (5)$$

Second law efficiency (Exergetic efficiency) is given as:

$$\eta_{II} = \frac{\eta_F}{\varepsilon_F}\left(\frac{R_{PH}+\varepsilon_p}{R_{PH}+1}\right) \qquad \ldots (6)$$

In the above the exergy factors are always less than one because there is degradation of energy always. The energy equation for different components in fig 3.1 is as follows:

TABLE3.1 ENERGY BALANCE EQUATIONS FOR GAS TURBINE COGENERATION WITH REHEAT

Component	Energy Equations
Air Compressor	Work done, $W_c = \dot{m}_a(h_2 - h_1)$ Efficiency, $\eta_c = \frac{h_{2s}-h_1}{h_2-h_1}$
Combustion chamber	$\dot{m}_f \times LHV = (\dot{m}_a + \dot{m}_f)\, C_{pg}\, \Delta T$ $= (\dot{m}_a + \dot{m}_f)\, C_{pg}(T_3 - T_2)$

Components	Energy Equations
Gas Turbine	Efficiency, $\eta_{T1} = \frac{h_4 - h_5}{h_4 - h_{5s}}$ $\eta_{T2} = \frac{h_5 - h_6}{h_6 - h_{6s}}$ Work done, $W_{T1} = (\dot{m}_a + \dot{m}_f)(h_4 - h_5)$ $W_{T2} = (\dot{m}_a + \dot{m}_f)(h_5 - h_6)$ Net work done, $W_T = W_{T1} + W_{T2}$
Heat Exchanger	Process Heat : $\dot{Q}_P = \dot{m}_s(h_g - h_c) = (\dot{m}_a + \dot{m}_f)\, C_P(T_6 - T_7)$ Exergy content of process heat: $\dot{B}_P = \dot{m}_s((h_g - h_c) - T_o(s_g - s_c)$ Exergy factor of process heat: $\dot{\varepsilon}_P = 1 - \left[\frac{To(s_g - s_c)}{(h_g - h_c)}\right]$

Where, h_g and h_c are the enthalpy of saturated vapour at process steam pressure and enthalpy of condensate return respectively. Now on the application of first law in heat recovery steam generator we get the following equation:

$$\dot{m}_s((h_g - h_f) = \dot{m}_a \left(1 + \frac{F}{A}\right)\left[\frac{h_6 - h_{PP}}{M}\right] \quad \text{.....(7)}$$

Now a ratio is obtained using the above equations:

$$\frac{\dot{Q}_P}{\dot{m}_a} = \left(1 + \frac{F}{A}\right)\left[\frac{h_6 - h_{PP}}{M}\right]\left(\frac{h_g - h_c)}{(h_g - h_f)}\right) \quad \text{.....(8)}$$

This ratio above represents the process heat given out by the unit air mass flow rate. It is compulsory to calculate of composition of exhaust gases for the above given quantity.

3.3.2 Thermodynamic modeling of the gas turbine with the inlet evaporative cooling

The design of system employing evaporative cooling is for a particular surrounding condition. It is not really the maximum ambient temperature along with least relative humidity that defines the maximum capacity of evaporative cooler. Basic fact is that the enhanced power is utmost requirement during the peak loading conditions; the selection of design point either based on this point or this point and the maximum capacity point combined together. The evaporating cooling system works by cooling air with the help of humidification in an adiabatic process. Also, during the process wet bulb temperature of air is seemed to be non-variant. The energy balance equations are used for the performance analysis of the gas turbine cogeneration cycle with inlet evaporative cooling. The equations are written taking reference from Santos [2012]. The important components of the inlet evaporative cooling system are analyzed under various process parameters in the thermodynamic modeling for its effective performance. These equations are written for the gas turbine cogeneration system as shown in figure 3.3. These are as follows:

Evaporative cooler effectiveness is given by:

$$\varepsilon = \frac{T_{2db} - T_{3db}}{T_{2db} - T_{2wb}} \qquad \ldots (9)$$

After cooling process inlet air temperature is calculated by formula given as:

$$T_3 = T_{b2} - \varepsilon\,(T_{b2} - T_{w2}) \qquad \ldots (10)$$

Evaporated mass flow rate of water in association with the evaporative cooling is given by:

$$\dot{m}_w = \dot{m}_a\,(\omega_2 - \omega_3) \qquad \ldots (11)$$

Cooling load associated with the evaporative cooling system is given by:

$$\dot{Q}_{cl} = \dot{m}_a\,C_{pavg}\,(T_2 - T_3) \qquad \ldots (12)$$

From the equations obtained for the gas turbine cogeneration system with reheat and the equations obtained for evaporative cooler. The two systems are thus combined to form the proposed system as shown in fig. 3.4. The equations involved in the analysis of the proposed system are discussed below in table 3.2.

TABLE 3.2 ENERGY EQUATIONS INVOLVED IN PROPOSED SYSTEM (fig 3.4)

Components	Equations
Evaporative cooler	Effectiveness, $\varepsilon = \frac{T_{2db}-T_{3db}}{T_{2db}-T_{2wb}}$ $T_3=T_{b2} - \varepsilon\,(T_{b2}-T_{w2})$ $\omega_3= \frac{C_{Pa}(T_{2db}-T_3)+\omega_2(C_{pv}\,T_{2db}+L-C_{pw}T_3)}{C_{pv}T_3+L+C_{pw}T_3}$ $\dot{m}_w = \dot{m}_a\,(\omega_2-\omega_3)$ Cooling load, $\dot{Q}_{cl} = \dot{m}_a\,C_{pavg}\,(T_2-T_3)$
Air Compressor	Work done, $W_c= \dot{m}_a(h_4-h_3)$ Efficiency, $\eta_c = \frac{h_{4s}-h_3}{h_4-h_3}$
Combustion Chamber	$\dot{Q}_{in1}= \dot{m}_a\,C_{pg\,,avg}\,((T_5-T_4)$ $\dot{m}_f = \frac{\dot{Q}_{in2}/LHV}{\eta_{combustor}}$ $\dot{m}_f \times LHV = (\dot{m}_a + \dot{m}_f)\,C_{pg}\,\Delta T$ $= (\dot{m}_a + \dot{m}_f)\,C_{pg}(T_5-T_4)$
Gas Turbine	HPT: $\eta_{T1}=\frac{h_4-h_5}{h_4-h_{5s}}$, $W_{T1}= (\dot{m}_a + \dot{m}_f)(h_5 - h_6)$ LPT: $\eta_{T2}=\frac{h_5-h_6}{h_6-h_{6s}}$, $W_{T2}= (\dot{m}_a + \dot{m}_f)(h_7 - h_8)$ $W_T = W_{T1}+W_{T2}$
Reheat Combustor	$\dot{Q}_{in2} = \dot{m}_a\,C_{pg\,,avg}\,((T_7-T_6)$ $\dot{m}_f = \frac{\dot{Q}_{in2}/LHV}{\eta_{combustor}}$

Heat Recovery steam Generator (HRSG)	Process Heat : $\dot{Q}_P = \dot{m}_s(h_g - h_c) = (\dot{m}_a + \dot{m}_f)\, C_P(T_8 - T_7)$ Exergy content of process heat: $\dot{B}_P = \dot{m}_s((h_g - h_c) - T_o(s_g - s_c)$ Exergy factor of process heat: $\dot{\varepsilon}_P = 1 - \left[\frac{To(s_g - s_c)}{(h_g - h_c)}\right]$

Various Performance parameters to be calculated are as follows:

Net work done from gas turbine is given by:

$$W_{net} = W_T - W_c \qquad .....(13)$$

Specific Fuel consumption is determined by:

$$SFC = \frac{3600\ \dot{m}_f}{W_{net}} \qquad .....(14)$$

Fuel-Utilization Efficiency/ First law efficiency(η_I) :

$$\eta_I = \frac{W_{el} + \dot{Q}_P}{H_F} \qquad .....(15)$$

Where W_{el} electrical power output which is given by:

$$W_{el} = \eta_g\ W_{net} \qquad ......(16)$$

Power to heat ratio (R_{PH}):

$$R_{PH} = \frac{Wel}{\dot{Q}_P} \qquad ....(17)$$

Second-law Efficiency (η_{II}):

$$\eta_{II} = \frac{W_{el}^{\cdot} + e_p}{e_f} \quad \text{…..(18)}$$

Where, exergy associated with the fuel(e_f) :

$$e_f = e_{f_{cc}} + e_{f_{reh}} \quad \text{….(19)}$$

3.4 MODEL VALIDATION:

The input process parameters are taken from the reference paper to validate the results for both gas turbine with inlet evaporative cooling and gas turbine cogeneration with reheat. The two models were validated individually and then integrated into single system of gas turbine cogeneration system with reheat and inlet evaporative cooling i.e. proposed system.

Model validation of reheat gas turbine cogeneration system:

The design and validation of the model, the basic input parameters taken for the system are given in table 3.3, the following input process parameters have same values as taken in A.Khaliq (2004) [19] and A.Khaliq and Kaushik (2007)[20]

TABLE 3.3 BASIC PARAMETERS DATA FOR THE CALCULATION OF RESULTS [19]

Parameters	Values
Cycle Pressure Ratio	34
Compressor Efficiency (η_c)	85
Air flow (Kg/s)	120
Turbine Inlet Temperature (˚C)	1150
Turbine Efficiency, η_T (%)	85
Pressure drop in Combustor (%)	3
Fuel	Natural gas
Excess Air Percent (%)	125

Parameters	Values
Ambient temperature (T_o), ˚C	20
Temperature of condensate return, T_c (˚C)	100
Number of reheats	1
Cycle efficiency (%)	37.3
Exhaust Temperature (˚C)	704
Exergy Factor of fuel, ε_f	1
Efficiency of combustion chamber (η_{cc}) ,%	95
Lower Heating Value (LHV) , MJ/kg	42

The power output and thermal efficiency value is taken from the table 3.3 and with these values the heat input , fuel input and mass flow of air can be calculated.

The following equation is combustion equation for complete combustion of any hydrocarbon with provided amount of theoretical air:

$$C_\alpha H_\beta + \lambda\left(\alpha + \frac{\beta}{4}\right)(O_2 + 3.76N_2) \longrightarrow \alpha CO_2 + \frac{\beta}{4} H_2O + 3.76\,\lambda\left(\alpha + \frac{\beta}{4}\right)N_2 + (\lambda\text{-}1)\left(\alpha + \frac{\beta}{4}\right)O_2$$

Where α is the number of carbon atoms in the fuel, λ is the percent of theoretical air and β is the number of hydrogen atoms in the fuel. Here $\left(\alpha + \frac{\beta}{4}\right)$ indicates the minimum number of moles that O_2 contains per mole of $C_\alpha H_\beta$.

The above equation is used for the calculation of the molecular weight (M) , the composition of the turbine exhaust and the fuel air ratio with the help of the data for excess air given in table 3.3. Now with the help of turbine exhaust composition ($h_6 - h_{pp}$) where h_{pp} is the enthalpy of gas mixture at pinch point. As a result, the process heat can be computed using the equation given for $\dot{Q}_p$ from table 3.1. The exergy factor ε_p can also be calculated from the equation given for it in table 3.1. After determining the above data we can now calculate energetic efficiencies (η_f, RPH) , an exergetic efficiency (η_{II}).

The cycle discussed above has been analysed and validated and following data is observed. Firstly the effect of pinch point temperature on various output parameters such as fuel utilization efficiency (η_f) , power to heat ratio (RPH) and on second law efficiency (η_{II}) with the help of EES software.

Table 3.4 Comparison between the reference work [19] and validated data for reheat cogeneration gas turbine cycle at 1MPa process steam pressure at variable pinch point temperature.

Temperature (˚C)	η_f (Reference work)	η_f (Validated data)	RPH (Reference work)	RPH (Validated data)	η_{II} (Reference work)	η_{II} (Validated data)
0	0.90	0.88	0.761	0.758	0.545	0.541
10	0.875	0.859	0.81	0.80	0.54	0.539
20	0.84	0.843	0.852	0.848	0.531	0.530
30	0.825	0.821	0.91	0.90	0.529	0.529
40	0.80	0.81	0.94	0.935	0.525	0.51
50	0.785	0.771	0.98	0.975	0.521	0.5

Also , the effect of the process steam pressure at 20˚C ambient temperature is analysed on the output parameters Temperature (˚C), η_f (Reference work), η_f (Validated data), RPH (Reference work), RPH (Validated data), η_{II} (Reference work), η_{II} (Validated data).

Table 3.5 Comparison between the reference work [19] and validated data for reheat cogeneration gas turbine cycle at 20˚C ambient temperature at variable process team pressure.

Pressure (MPa)	η_f (Reference work)	η_f (Validated data)	RPH (Reference work)	RPH (Validated data)	η_{II} (Reference work)	η_{II} (Validated data)
0	0.88	0.876	0.80	0.78	0.49	0.489
0.5	0.87	0.86	0.84	0.82	0.50	0.51
1	0.856	0.86	0.86	0.85	0.513	0.52
1.5	0.848	0.839	0.89	0.89	0.543	0.539
2	0.84	0.833	0.92	0.91	0.554	0.55

2.5	0.832	0.829	0.95	0.94	0.561	0.558
3	0.822	0.82	0.97	0.96	0.57	0.568
3.5	0.81	0.81	0.99	0.98	0.582	0.58

Model validation of the gas turbine with inlet evaporative cooling:

The following input process parameters were taken from the reference paper Santos [12] so as to validate the results and check the equation for the inlet evaporative cooling process. The cycle is single shaft, simple industrial engine

TABLE 3.6 INPUT PRESSURE PARAMETERS F0R SELECTED GAS TURBINE [12]

Parameters	Values
Pressure Ratio	11
Turbine Inlet temperature (K)	1385
Air flow rate (Kg/s)	141
Isentropic efficiency of compressor (%)	85.4
Isentropic efficiency of turbine (%)	86.8
Combustion efficiency (%)	99
Inlet pressure laws (mm,H_2O)	100
Exhaust pressure laws (mm,H_2O)	200
Combustion chamber pressure laws (%)	1.2
Fuel	Natural Gas
LHV (MJ/kg)	48.235

The above values along with the effectiveness of evaporative cooler ε = 0.9 is used for the calculation of the validated results. Using the equation described for the inlet evaporative cooling, i.e. from equation (9) to equation (11) ,the work output and the thermal efficiency is calculated at relative humidity ϕ=18% and ϕ=60%. The comparison between the reference work results and validated results obtained by using EES software is in the table 3.7.

Table 3.7 Variation of power output with the intake air temperature

	ϕ=18%		ϕ=60%	
Intake air temperature (˚C)	Power output (Reference Data)	Power output (Validated Data)	Power output (Reference Data)	Power output (Validated Data)
8	40.5	40	39.5	38
16	38	39	37	37.4
24	36	36.4	34.8	34
32	34	35.1	32.5	33.1
40	32.2	31.4	30.5	31

.

Table 3.8 Variation of thermal efficiency with the intake air temperature

	ϕ=18%		ϕ=60%	
Intake air temperature (˚C)	Thermal efficiency (Reference Data)	Thermal efficiency (Validated Data)	Thermal efficiency (Reference Data)	Thermal efficiency (Validated Data)
8	30.35	30.25	30.1	30
16	29.9	28.45	29.4	29.2
24	29.4	29	28.75	28
32	28.9	28.4	28.1	26.8
40	28.4	27.2	27.3	27.6

So, from the model validation, we can conclude that there is a similarity in the results of the reference work and the result obtained from EES software. The data is approximately similar to that of the reference work data and the nature of variation is also the same, so we can move forward with the equations used in the thermodynamic modeling to obtain results for the proposed system. The results for the proposed system are discussed in Chapter 4.

CHAPTER 4

RESULTS AND DISCUSSION

On the basis of thermodynamics analysis the results have been obtained and presented in this chapter for studying the effect of various input parameters such as turbine inlet temperature, compressor inlet temperature, compression ratio, ambient temperature on gas turbine performance parameters i.e. thermal efficiency or first law efficiency, second law efficiency and specific work output.

4.1 VARIATION OF DIFFERENT OUTPUT PARAMETERS WITH AMBIENT INTAKE TEMPERATURE

4.1.1 Variation of Cooling load with ambient Temperature

From Fig 4.1 it could be depicted that total cooling load increases with increasing ambient temperature. However, we know the cooling load is quite low as compared to absorption chillers or mechanical chillers and it is also quite steady and smooth. The parameters are humidity ratio (ϕ=18%) and turbine inlet temperature 1380 K are constant. The evaporative cooler has the efficiency ε=0.9. The cooling load varies from 1MW to 1.5MW when the cooler intake temperature varies from 275 K to 315 K.

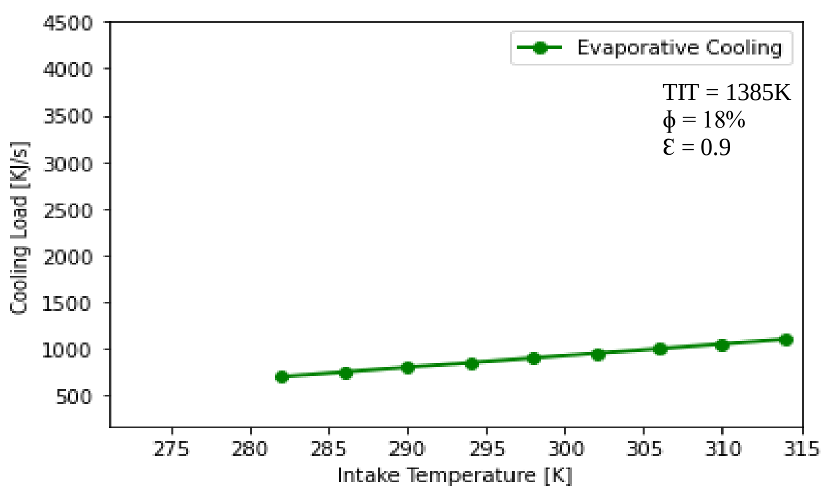

Fig 4.1 Cooling load variation with Intake air Temperature

4.1.2 Variation of Evaporated mass flow rate with ambient Intake air temperature

Fig. 4.2 depicts the effect of the ambient temperature on the evaporated mass flow rate of water. As per previous studies the evaporative cooling has better performance at lower humidity. The turbine inlet temperature is 1380K which is constant as the cooling load is limited by the wet bulb temperature. Therefore, figure 4.1 and 4.2 are plotted at relative humidity ϕ =18% showing the variation of cooling load and mass flow rate. The mass flow rate varies from the 0.35 kg/s to 0.75 kg/s as the temperature varies from 275K to 315K.

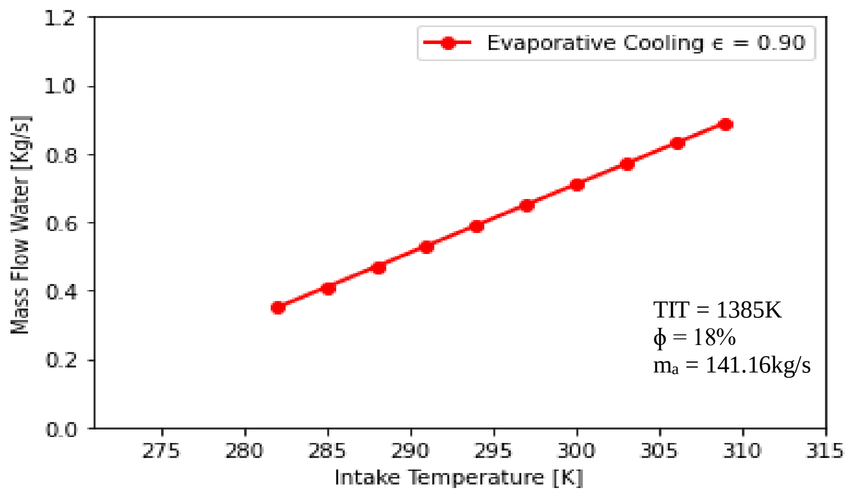

Fig 4.2 Mass flow rate variation with Intake Temperature

4.1.3 Variation of work output with ambient intake temperature

The work output decreases with increase in the ambient intake temperature as depicted in the fig.4.3. The evaporative cooler has the efficiency of 0.9 and the turbine inlet temperature is 1385 K and the relative humidity is ϕ= 18%. There is a steep drop with the increase in the ambient intake temperature. As in the figure 4.3 the power output decreases from 283 KJ/kg to 230 KJ/kg on varying the temperature from 275K to 315K.

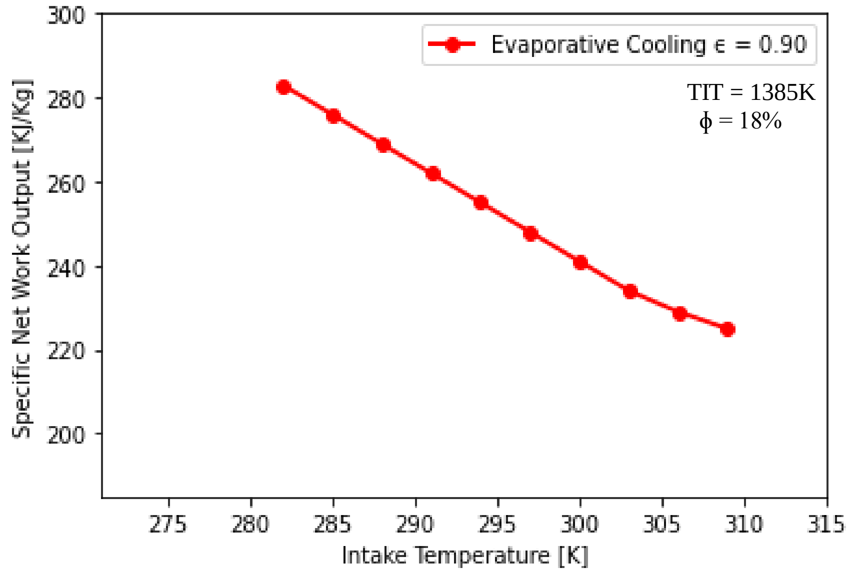

Fig 4.3 Specific Net work output variation with Intake air Temperature

4.1.4 Variation of thermal efficiency with Ambient Intake Temperature

Fig 4.4 depicts the variation of thermal efficiency with Ambient Intake temperature and it can be seen clearly that thermal efficiency decreases as the ambient temperature increases because thermal efficiency is ratio of net work output which is work produced by the turbine minus the compressor work and heat supplied in combustion chamber. The plot is made at constant turbine inlet temperature of 1385K a constant amount of fuel is supplied in combustion chamber which means thermal efficiency is directly proportional to net specific work output and net specific work output is decreases with increase in ambient temperature so thermal efficiency also decreases with increase in ambient temperature. When ambient temperature increases from 275K to 315K, thermal efficiency drops approximately by 1.8%.

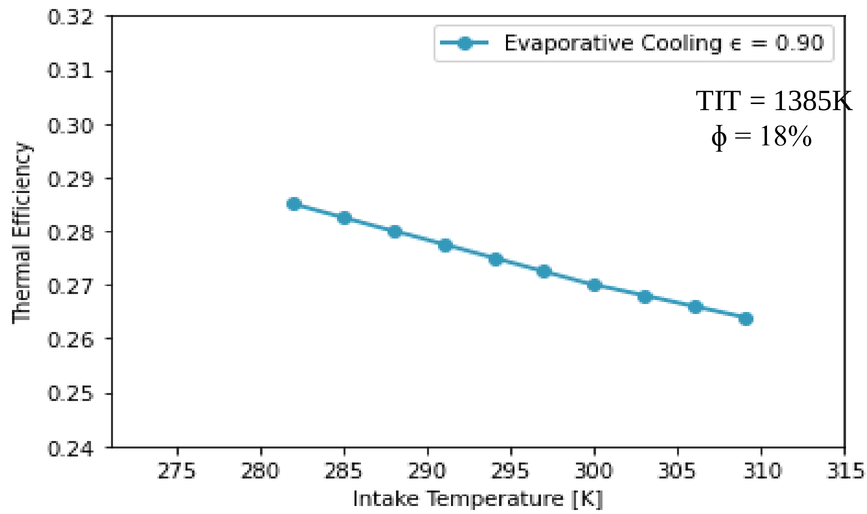

Fig 4.4 Thermal efficiency variation with Intake air Temperature

4.1.5 Variation of thermal efficiency with intake temperature for ϕ=18% and ϕ=60%

Figure depicts the variation of the thermal efficiency at two relative humidity values i.e. ϕ=18% and ϕ=60%. So, from the graph it is clear that at lower humidity thermal efficiency is higher as compared to that at the higher relative humidity. However, in both the cases thermal efficiency decreases with the increase in the intake temperature and drops 1.4 % in case of relative humidity 60% and 1.04% drop in case of relative humidity equal to 18%.

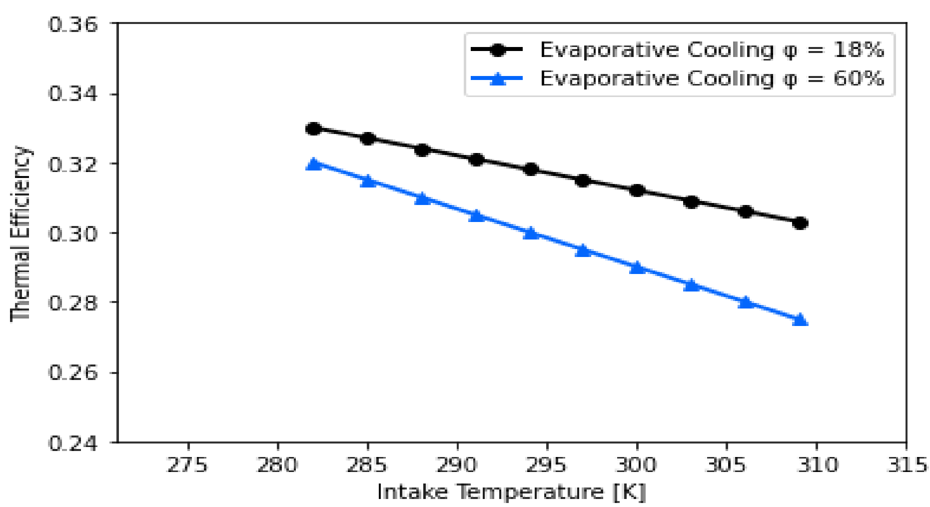

Fig 4.5 Thermal efficiency variation with Intake air Temperature

4.1.6 Variation of work output with intake air temperature for ϕ=18% and ϕ=60%

Figure depicts the variation of the work output at two relative humidity values i.e. ϕ=18% and ϕ=60%. So, from the graph it is clear that at lower humidity work output is slightly higher as compared to that at the higher relative humidity. However, in both the cases the work output decreases with the increase in the intake temperature.

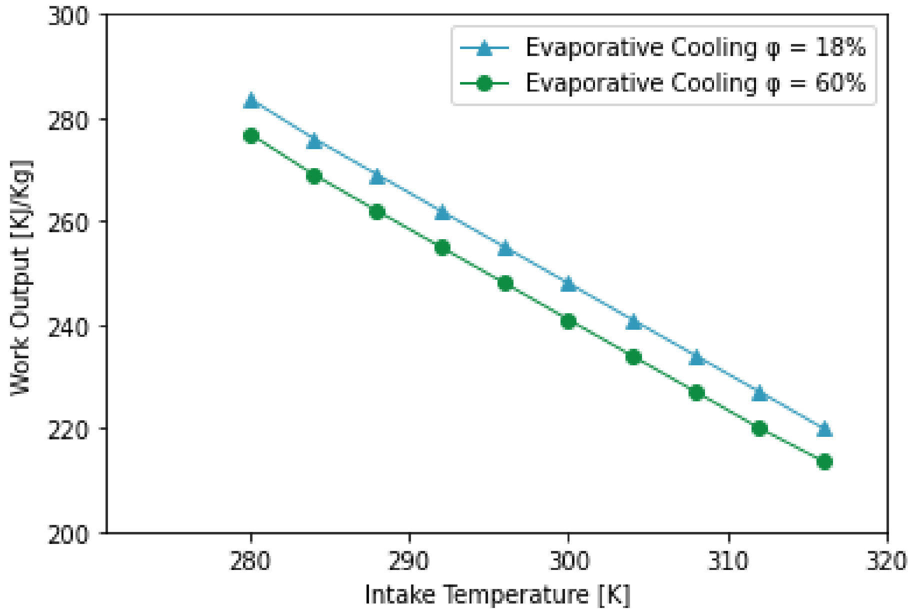

Fig 4.6 Work Output variation with Intake Temperature

4.2 VARIATION OF OUTPUT PARAMETERS WITH RELATIVE HUMIDITY

4.2.1 Specific Net Work Output V/s Relative Humidity

We can clearly see from the Fig 4.7 that the plot between the specific net work output and relative humidity that the specific work output tends to decrease as the humidity increases. So, lower the relative humidity, greater is the specific work output. The reason behind it is that when ambient relative humidity increases then wet bulb temperature also increases which means exit temperature of evaporative cooler increases, so specific net work output decreases. Here, there is decrease of approximately 60.8 KJ/kg as the relative humidity drops from 0.1 to 1.

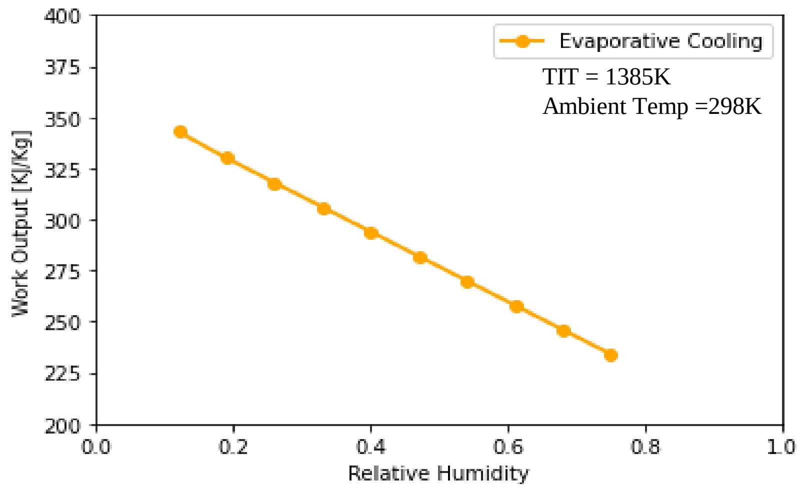

Fig 4.7 Work output variation with relative humidity

4.2.2 Thermal Efficiency v/s Relative humidity

Fig 4.8 depicts the variation of thermal efficiency with ambient relative humidity .It states that thermal efficiency decreases with increase in ambient relative humidity .The reason behind this is that an increase in ambient relative humidity increases the wet bulb temperature also which means exit temperature of evaporative cooler increases. Thus, for a particular compressor inlet temperature, the compressor work which decreases the net specific work output. We know that thermal efficiency is the ratio of net specific work output and heat added in combustion chamber, for particular TIT heat added in combustion chamber is constant so for this case thermal efficiency is directly proportional to net specific work done so with decrease in net specific work output thermal efficiency also decrease. Thermal efficiency drops from 0.37 to 0.33, when ambient relative humidity increases from 0.1 to 1.

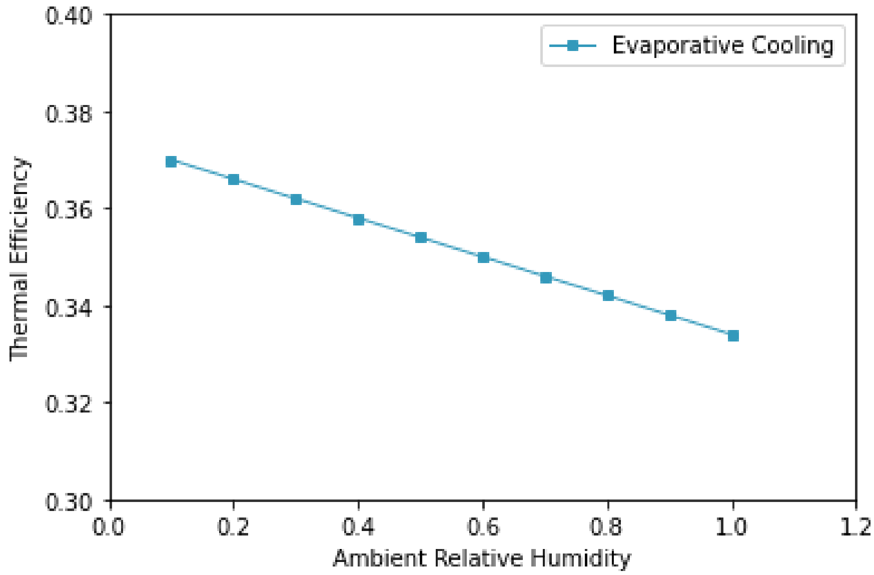

Fig 4.8 Thermal Efficiency Variation with Relative Humidity

4.3 VARIATION IN OUTPUT PARAMETERS WITH TURBINE INLET TEMPERATURE

4.3.1 Effect of Turbine Inlet Temperature on Power to heat ratio

Figure 4.9 depicts the variation of power heat ratio with the turbine inlet temperature for the proposed cycle that is reheat cogeneration cycle with the inlet evaporative cooling. We can clearly see that as the turbine inlet temperature increases from 1200K to 1900K, power to heat ratio decreases approximately from 18 to 8.

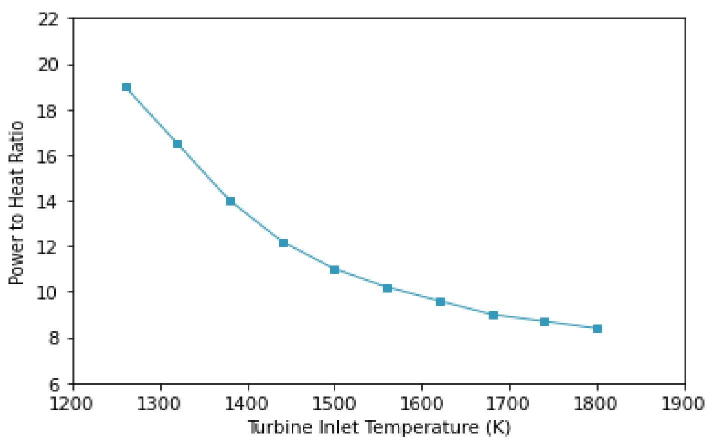

Fig 4.9 Power to heat ratio variation with Turbine Inlet Temperature

4.3.2 Effect of Turbine Inlet Temperature on First Law Efficiency

In the Figure 4.10 it is depicted from the graphs that the variation of thermal efficiency for three cycles is represented with the Turbine Inlet Temperature (TIT) .It shows that for the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate constant there is slight difference in the efficiency of the reheat cogeneration cycle and reheat cogeneration cycle with inlet evaporative cooling in gas turbine. However, it is clear that simple gas turbine with inlet evaporative cooling has lesser efficiency.

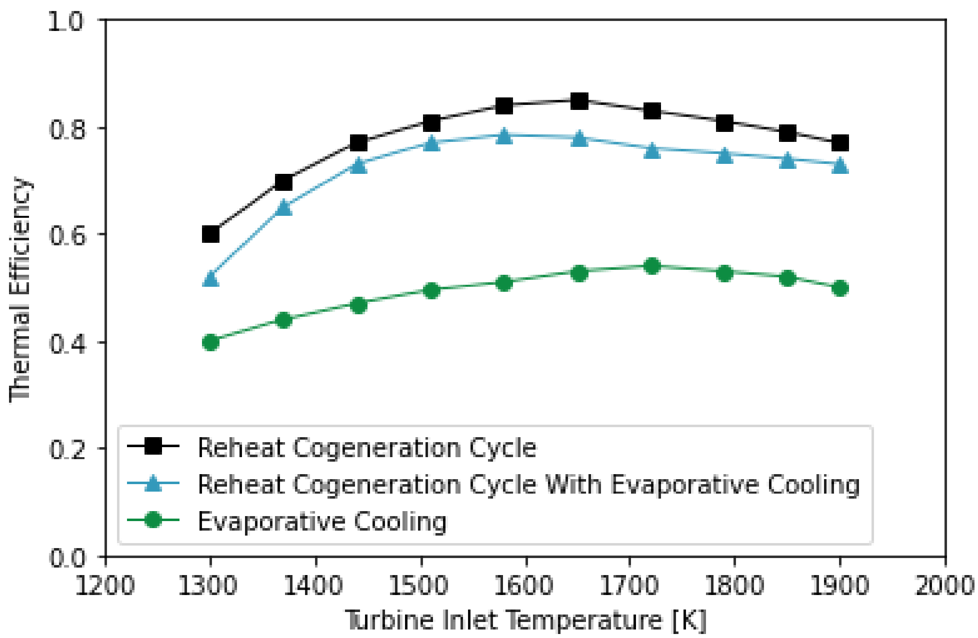

Fig 4.10 Thermal or first law efficiency variation with Turbine Inlet Temperature

4.3.3 Effect of Turbine Inlet Temperature on Second law efficiency

Fig 4.11 shows the effect of turbine inlet temperature on the second law efficiency for t he proposed cycle i.e. reheat cogeneration with inlet evaporative cooling in gas turbine. The turbine inlet temperature increases at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate constant. Second law efficiency is taken in percentage and there is increase in efficiency from 42% to 77% as the turbine inlet temperature increases from 1200K to 1900 K .

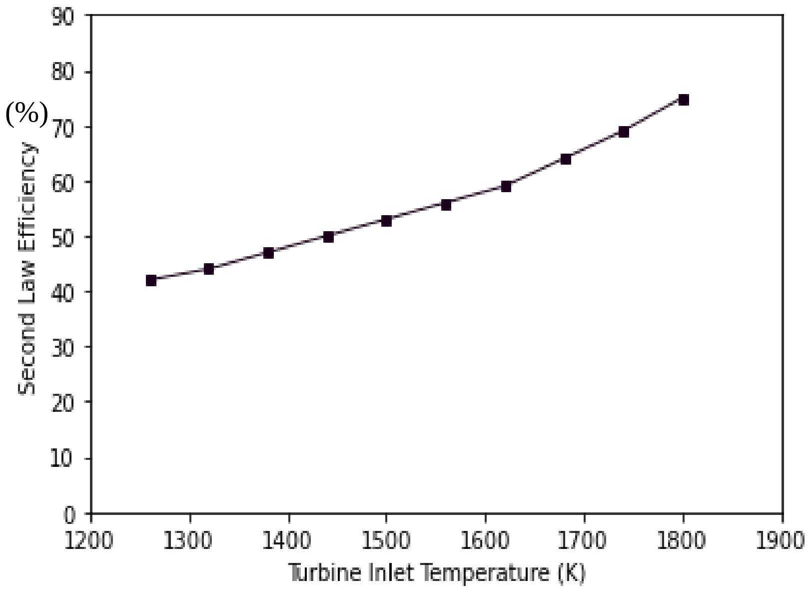

Fig 4.11 Effect of Turbine Inlet Temperature on Second law efficiency

4.4 VARIATION OF OUTPUT PARAMETERS WITH COMPRESSION RATIO

4.4.1 Variation of Power output with compression ratio

Fig 4.12 shows the effect of Compression ratio on the work output having values in KJ/kg for the proposed cycle i.e. reheat cogeneration with inlet evaporative cooling in gas turbine. As the compression ratio varies at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate constant, there is decrement in the work output. So, from the figure we can see that as the compression ratio increases from 5 to 35 there is the drop in the net work output and the drop is equal to 220 KJ/kg approximately.

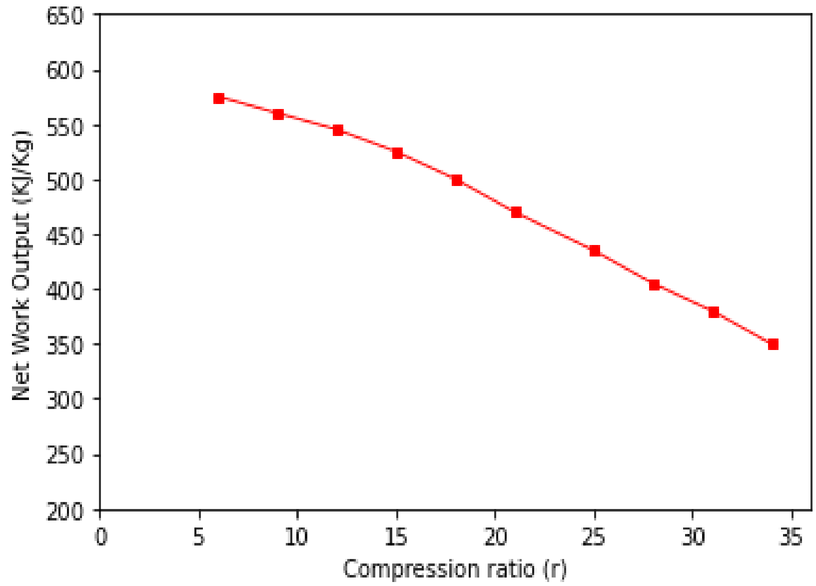

Fig.4.12 Effect of Compression Ratio on Net Work Output

4.4.2 Variation of First law efficiency with compression ratio

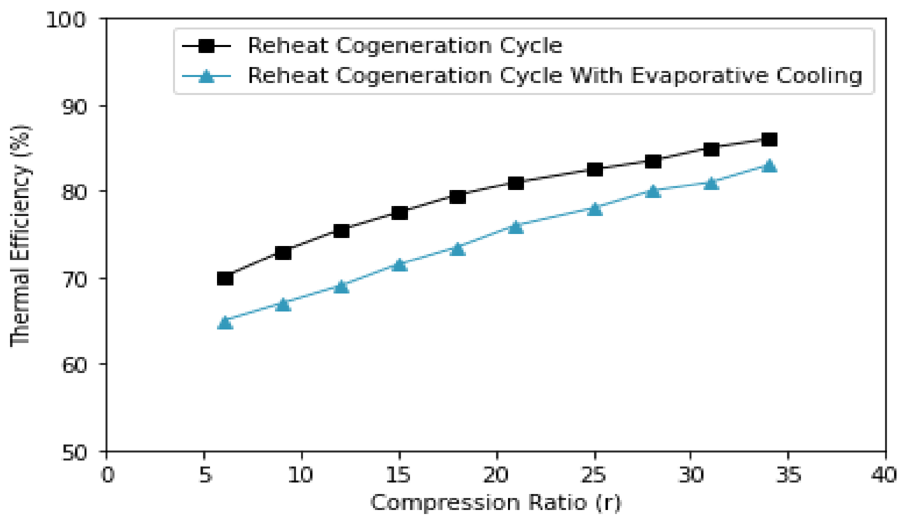

Fig 4.13 Effect of compression ratio on Thermal Efficiency

The figure 4.13 shows the effect of compression ratio on the thermal efficiency for the proposed cycle i.e. reheat cogeneration with inlet evaporative cooling in gas turbine as well as reheat cogeneration in gas turbine. As the compression ratio varies at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate and the turbine inlet temperature constant. As the compression ratio varies from 5 to 40 the efficiency increases for both the cycles. However, the reheat cogeneration cycle has higher efficiency as compared to the proposed cycle as seen in the above graphical representation.

4.4.3 Variation of Second law efficiency or exergy efficiency with Compression ratio

The figure 4.14 shows the effect of compression ratio on the second law efficiency for the proposed cycle that is reheat cogeneration with inlet evaporative cooling in gas turbine. As the compression ratio varies at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate and the turbine inlet temperature constant. As the compression ratio varies from 5 to 40 second law efficiency increases approximately from 73.1% to 85%.

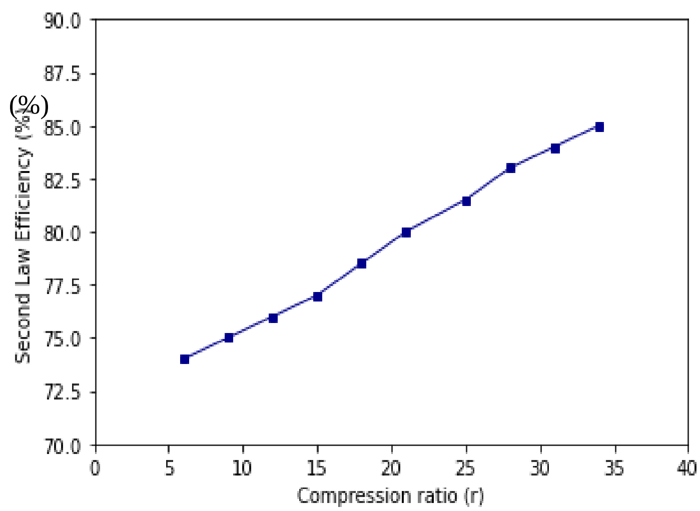

Fig 4.14 Effect of compression ratio on Second law efficiency

4.4.4 Process heat variation with Pressure ratio

The figure 4.13 shows the effect of compression ratio on the process heat for the proposed cycle i.e. reheat cogeneration with inlet evaporative cooling in gas turbine. As the compression ratio varies at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate and the turbine inlet temperature constant. As the compression ratio varies from 5 to 40 process heat increases significantly as can be seen in the figure 4.15.

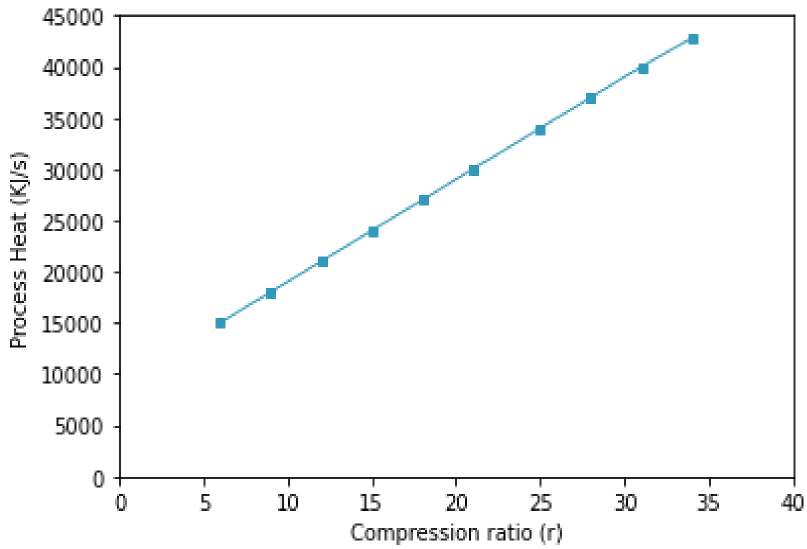

Fig 4.15 Effect of compression ratio on process heat

4.5 VARIATION OF OUTPUT PARAMETERS WITH PINCH POINT TEMPERATURE

4.5.1 Power to heat ratio v/s Pinch Point Temperature

The figure 4.16 shows the effect of pinch point temperature on the power to heat ratio for the proposed cycle i.e. reheat cogeneration with inlet evaporative cooling and reheat cogeneration in gas turbine. As the compression ratio varies at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate and the turbine inlet temperature constant. As pinch point varies from 10˚C to 70˚C power to heat ratio increases for both the cycle. However, the reheat cogeneration cycle with evaporative cooling as compared to reheat cogeneration cycle has higher efficiency as compared to the proposed cycle as can be seen in the figure and the margin is 0.2.

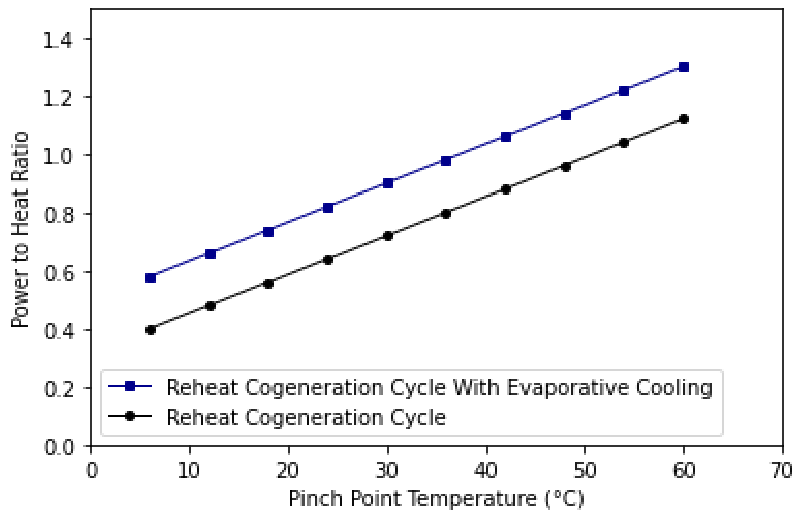

Fig 4.16 Effect of Pinch Point Temperature on Power to Heat Ratio

4.5.2 First Law Efficiency v/s Pinch Point Temperature

The figure 4.13 shows the effect of pinch point temperature on the thermal efficiency for the proposed cycle i.e. reheat cogeneration with inlet evaporative cooling in gas turbine as well as reheat cogeneration in gas gas turbine. As the compression ratio varies at the constant ambient temperature 298 K and 18% relative humidity keeping the mass flow rate and the turbine inlet temperature constant. As the compression ratio varies from 3.5 to 8 the efficiency decreases for both the cycles. However, the reheat cogeneration cycle has marginally higher efficiency as compared to the proposed cycle that can be seen in the figure.

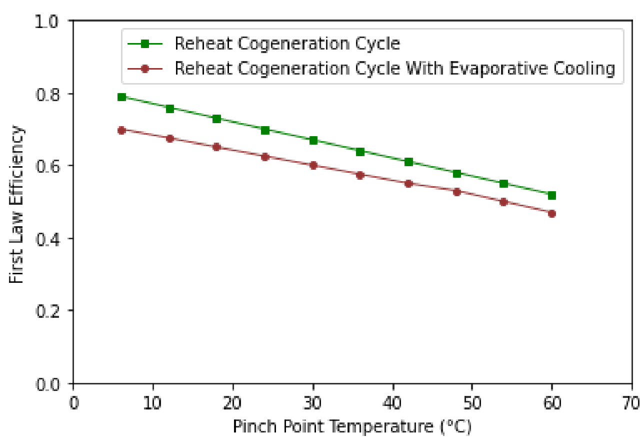

Fig. 4.17 Effect of pinch point temperature on First Law Efficiency

CHAPTER 5

CONCLUSION AND FUTURE WORK

As per the results and discussion presented in the chapter 4 it can be concluded that the efficiency calculation for the gas turbine cogeneration with the reheating and inlet evaporative cooling thermal efficiency is greater than that of the simple gas turbine with inlet evaporative cooling but slightly lower than the Gas Turbine Cogeneration System with Reheat. The variation of the Specific work output, Thermal efficiency and Power to heat ratio with the turbine inlet temperature and the compression ratio is quite significant. The variation of thermal efficiency and power output with pinch point temperature is seen to linearly vary. The proposed cogeneration cycle with the inlet evaporative cooling and reheat can be further improved and analyzed for the cost analysis as well as optimized using the optimization techniques.

The gas turbine cogeneration systems applications are continuously growing as it is considered that there are various techniques such gas turbines can be effectively and economically utilized in the respective plants. However, there are so many advantages but some disadvantages due to higher number of components employed and more cost associated with them. So, a more energy efficient way of producing energy can be used combining the renewable sources of energy for the cogeneration. Biomass energy and solar energy being the topmost choice for the cogeneration purpose instead of fuel used. However, wind energy and other non -conventional sources can also be used by finding effective way to harness the energy.

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