COMPARATIVE STUDY OF GAS TURBINE-HRSG-ORC AND GAS TURBINE-HRSG-SUPERCRITICAL ORC COGENERATION SYSTEMS THROUGH ENERGY AND EXERGY ANALYSIS

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CANDIDATE'S DECLARATION

I, Priyanshu Thakur, 2K17/THE/10 student of M.Tech (Thermal Engineering), hereby declare that the project Dissertation title "Comparative Study of GT-HRSG-ORC and GT-HRSG-Supercritical ORC Cogeneration Systems through Energy and Exergy analysis" 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 the Project Dissertation titled "Comparative Study of GT-HRSG-ORC and GT-HRSG-Supercritical ORC Cogeneration Systems through Energy and Exergy analysis" which is submitted by Priyanshu Thakur, 2K17/THE/10, Mechanical Engineering Department, 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.

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ABSTRACT

The comparative study between the ORC and supercritical ORC has been under study in recent time and the comparatively enhanced performance of the supercritical ORC has made a good alternative to the subcritical ORC systems. In this work, the relative feasibility of the supercritical ORC over ORC was studied when they are positioned as the bottoming cycle in a cogeneration system. A comparative study was carried out to analyse two cogeneration systems, first consisting of Gas Turbine- HRSG-ORC cycle and second consisting of Gas turbine-HRSG-Supercritical ORC cycle. The comparative study was carried out though the energy and exergy analysis of the two systems. Though the energy and exergy analysis it was found that in the bottoming cycle supercritical ORC does performs better in compared to the subcritical ORC. The supercritical ORC register an increase in energy and exergy efficiency by 0.6 to 1.2 percentage point depending upon the value of the parameter taken under the parametric study. However, this comes at the expense of comparatively higher exergy destruction observed in the bottoming supercritical cycle up to certain levels of the parameter. The result also establishes that throughout the parametric range the work output of the bottoming supercritical ORC remains considerably higher than the subcritical ORC cycle.

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Abbreviations

PVR	Pressure Reduction Valve
WHRB	Waste Heat Recovery Boiler
HRSG	Heat Recovery Steam Generator
GT	Gas Turbine
T/TUR	Turbine
C/COMP	Compressor
SOFC	Solid Oxide Fuel Cell
HE /HX	Heat Exchanger
COMB/CC	Combustion Chamber
HTHX	High Temperature Heat Exchanger
LTHX	Low Temperature Heat Exchanger
HPG	High Pressure Generator
LPG	Low Pressure Generator
ABS	Absorber
CON	Condenser
EVA	Evaporator
CCHP	Combined Cooling Heating And Power
ORC	Organic Rankine Cycle
Sc. ORC	Supercritical ORC
APH	Air Pre Heater
LHV	Lower Heating Value Of Fuel
TIT	Turbine Inlet Temprature
EDR	Exergy Destruction Ratio
eff1	First Law Efficiency
Ex_eff	Exergy Efficiency

Symbols

η	Efficiency
ε	Effectiveness
r	Compression Ratio
COP	Coefficient Of Performance
$\dot{W}$	Work Output
$\dot{Q}$	Heat Flow
h	Enthalpy
s	Entropy
v	Specific Volume
p	Pressure
$\dot{m}$	Mass Flow Rate
$\dot{E}$	Exergy
$\dot{I}$	Exergy Destruction
τ	Air Fuel Ratio
P	Power Output

Subscripts

p	Pump
f	Fuel
is/s	Isentropic
wf	Working Fluid
D	Destruction
hsi	Heat Source Inlet
hso	Heat Source Outlet
csi	Cooling Source Inlet
cso	Cooling Source Outlet
gen	Generator

C	Condenser
CV	Control Volume
I	Inlet
E	Exit
tot	Total

CHAPTER 1

INTRODUCTION

1.1 COGENERATION SYSTEM

Cogeneration systems are defined as that energy producing system which produces more than two form of energy from the same fuel source. According to the World alliance of Decentralisation Energy, cogeneration system are defined as, “Process of producing both electrical and usable thermal energy (heating or cooling) at high efficiency and near the point use” [1] where as The Bureau of Energy Efficiency India defines cogeneration as "Sequential generation of two different form of energy from a single primary source, typically mechanical and thermal energy." [2]

1.2 SCENARIO OF COGENERATION SYSTEM

Cogeneration system dates back to 1880 when electricity was not popular as a primary source of energy instead steam was used as a major source for any of the activity ranging from heating a space to running a automobile and trains through steam engines. Those days electricity was merely emerging as a source of energy. With evolution of central power grid electricity emerged as primary source of energy and the utilisation of other form of energy started to decline, cogeneration system saw a decline in its utilisation around 1974 it accounted for only 4% of the electricity generation [3] . However when the oil crisis of 1973 crippled all the major developed countries the need for utilisation of systems like cogeneration system was felt. As a result of the sharp accrual in energy prices and decreased certainty of fuel supply supplies, engineers started to find new avenues to augment to their electricity generation by developing more efficient system that can utilise fuel in a rather distinctive manner like cogeneration. This lead to rejuvenation of the cogeneration system. However in recent years the cogeneration system has seen significant advancement, the reason for it is attributed to increasing shortage of the natural resources and slow pace in development of alternative sources of energy. Cogeneration system provides with increased efficiency of the system which helps in more efficient utilisation of the limited natural resources. While the conventional system provides with an efficiency of 30 - 40%, the integration of cogeneration system can enhance the efficiency up to 80% [4]. As a result of enhanced efficiency the cogeneration system has seen an increased utilisation in all the spheres, it finds potential application in industries like chemical and manufacturing industries, in buildings like hospitals as well as in IC engines systems. It has most positive effect on the industries, as cogeneration system provides the industries with an opportunity to enter the electricity generation and sales market. Cogeneration systems makes impact on both at micro and macro level, while at micro level, the one using such system will find a significant trimming down of their electricity bill, especially at the places where there is concurrent demand of electricity and heat, and a reasonable energy tariff is present in the country. On the other hand at macro level, it permits the private sector to bear a portion of financial burden of already laden state power utility grid. In addition to that, conservation and efficient utilisation of the state resources takes place [2]. The analysis of country wise utilisation shows that as of now cogeneration system contributes to around 11% of total power generation in European Union [5]. Till 2010 Brazil has a installed capacity of power generation around 12GW through cogeneration systems according to Association of Cogeneration Producer. Malaysia has been leading producer of electricity through cogeneration system, cogeneration system has been operational in Malaysia for around a decade i.e., since 2007, when its installed capacity was 713.2MW and is still flourishing [6]. There has been continuous increase in the utilisation of cogeneration system and there has been several work carried out on various cogeneration system prominent among them are steam turbine cogeneration, gas turbine cogeneration, reciprocating engine cogeneration system. System like fuel cell cogeneration and engine based cogeneration has been proved very efficient in its utilisation at residential and commercial building. However, currently only internal combustion engine based system has been proven to be cost effective.

1.3 NEED FOR COGENERATION SYSTEM

World energy scenario shows that there has been consistent rise in the demand for energy with rise in population of the world and with more and more countries increasing the pace of their growth. With limited natural resources available on earth and slow pace of development of renewable and nonconventional sources of energy, there has been an increasing need of managing the available resources in sustainable and efficient manner so that the life time of the available sources can be increased, to handle such a scenario cogeneration system has been evolving as a suitable alternative for efficient and effective utilisation of available resources. 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 utilisation efficiency of the fuel used up to 75% [7]. 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 [8]. 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 [9]. 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.

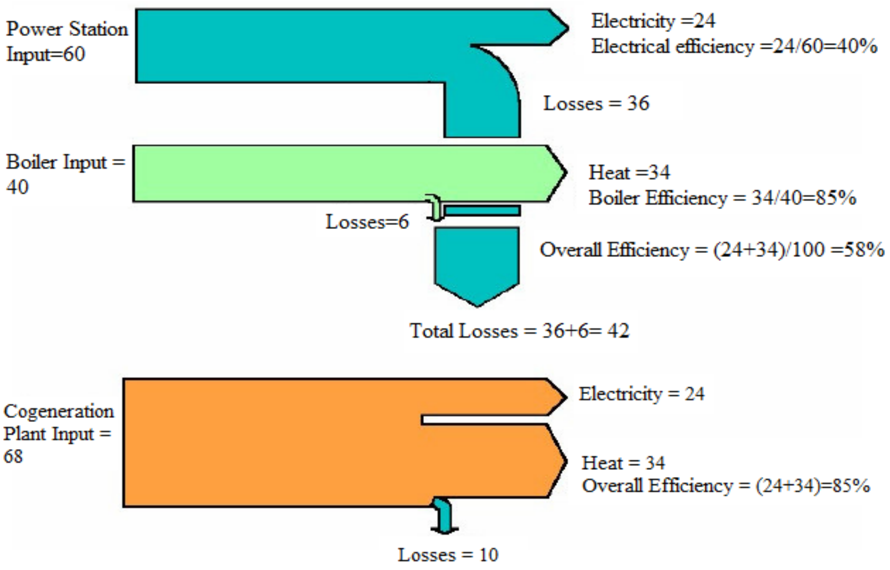


Figure 1.1: Cogeneration plant (Bottom) compared with conventional system (top) (Source: Ref. [2])

Gas turbine cogeneration system will provide the required assistance by significantly improving the percentage of thermal energy utilisation in the different industrial sectors. Figure 1.1 shows a general comparison of cogeneration system with respect to other conventional power plant and depict the how overall efficiency of utilisation of fuel can be enhanced through such system.

1.4 TYPES OF COGENERATION SYSTEM

Although there could be multiple cogeneration systems with different configuration, while considering various types of cogeneration system, some of these configurations are highly accepted among various industries due to the ease in its commercial availability and proven track record. Some of these highly accepted systems are as follows:

1. Steam turbine based cogeneration system.
2. Gas turbine based cogeneration system
3. Reciprocating engine based cogeneration system
4. Combined steam/gas turbine based cogeneration system

1.4.1 Steam turbine based cogeneration system

This system has the basic working concept of Rankine Cycle. In Rankine cycle, boiler act as the heat source in the system where steam is generated at high pressure and predetermined parameters by firing the fuel in the boiler. The highly pressurised steam thus produced is then expanded to low pressure stream through a turbine and produce mechanical power or electricity. The exhaust steam thus obtain in the system consist of heat energy which can be utilised further as a heat source in other industries such as in chemical industry to facilitates a chemical reaction. Cogeneration system is also used through extraction of steam through turbine. The extracted steam can be directly used as feed to process consumer or can also be used in hating the feed water there by increasing the efficiency of the overall cycle. It has been discovered that the non condensing steam turbine based cogeneration system has been the most efficient system used to meet the heat requirement of chemical processes in chemical industries. Figure 1.2 shows the configuration of non condensing steam turbine cogeneration system. This system has the ability to revamp the global efficiency up to 80%-85%. Various cogeneration system has different parametric requirement at which steam is required to be delivered this is accomplished by using different steam turbine. Various factor such as the required temperature or pressure of process steam, load condition on power generating unit power and steam demand variation etc contributes as a determining factor in selection of steam turbine for the process. Currently these systems are operation with variety of fuel types, they can be conventional fossil fuel based system such as coal based, oil based or natural gas based system or they can be non conventional fuel based systems such as plants operating on bio gas, bagasse or municipal waste. As a result the system is considered as highly flexible with respect to the type of fuel used. In spite of having certain advantages the it is considered suitable only for large scale system only and are not recommended for small scale systems as such system tends to become expensive and maintenance oriented.

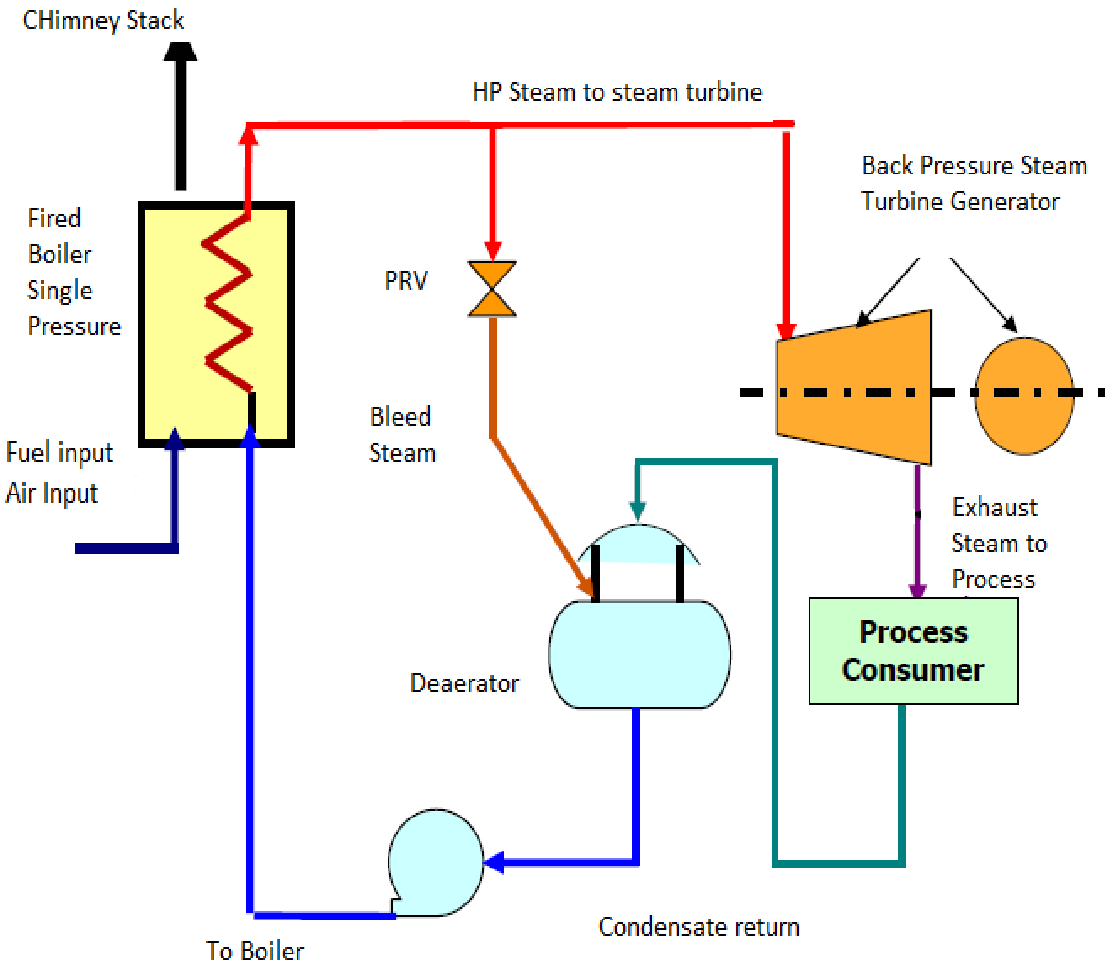


Figure 1.2: Steam turbine based cogeneration system (Source : Reference [2])

1.4.2 Gas Turbine based cogeneration system

These system works under the thermodynamic concept of Brayton cycle. In these kind of system the air is hauled from the atmosphere and is compressed to desired pressure. This high pressure air is then supplied into the combustion chamber, at which the fuel and pressurised air are mixed in desired proportion and the air fuel mixture is then ignited in the combustion chamber. The burning of the mixture produces combustion gasses at high pressure and temperature. This combustion gasses are then expanded through the gas turbine. The mechanical power obtained from the turbine is used to run generator and a part of it is used to run the air compressor. The fuel gasses obtained from the exhaust are generally at high temperature of 480-540 ℃. This high temperature air has the potential to be used as an heat source for production of process steam and to heat the air or as a heat source in other industries to provide heat for chemical reaction. Figure 1,3 represents a typical gas turbine based cogeneration system.

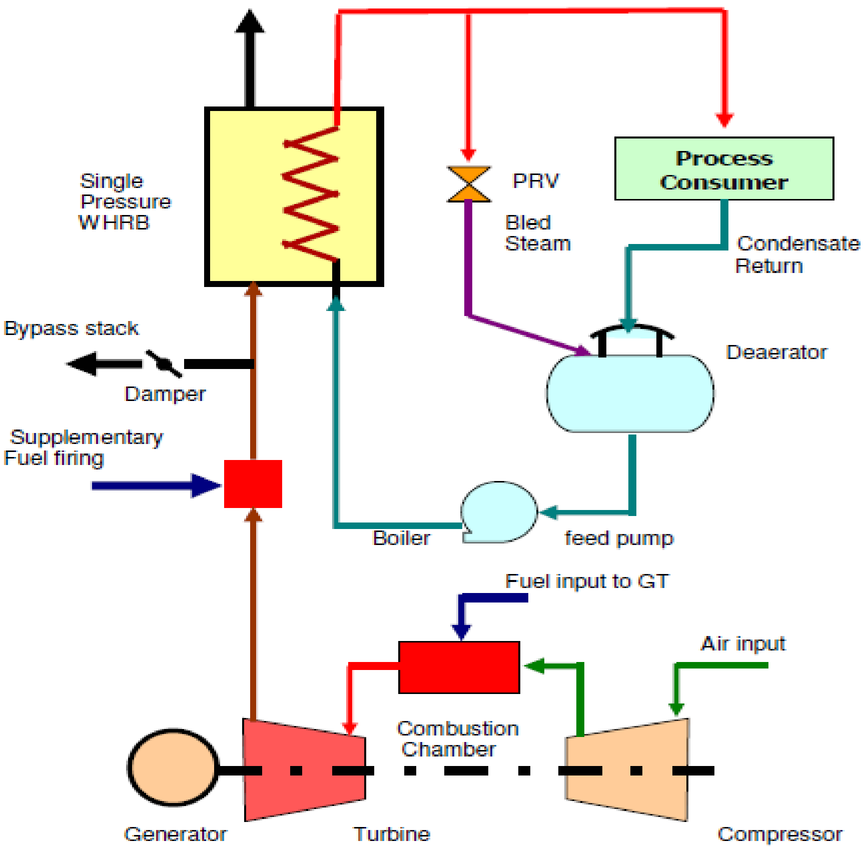


Figure 1.3: Gas turbine based cogeneration system (Source: Reference [2])

Generally the gas turbine based power plants in the industries are used produce electric power only nad has no secondary system of heat recovery. These plants generally attains an efficiency between range 24%-35% but with integration of heat recovery system such as heat recovery steam generator (HRSG) or waste heat recovery boiler (WHRB) and enhanced efficiency of 85-90% can be achieved. In alternate to that, the high temperature exhaust gasses can be used also for heating of water or providing hot air (Direct heating process) instead of generating steam. Chemical industries produce a consistent demand for steam as a result gas turbine based cogeneration system, producing process steam, are considered ideal for the chemical industries as it help in meeting the demand for electricity and steam simultaneously instead of simple steam generating system. Gas turbine cogeneration system provides with good performance with clean fuels like natural gas having no or very less ash content. Fuels like Naphtha, High Speed Diesel etc which are prominently known for their low ash content are also used in gas turbine cogeneration systems, Although the system has significant advantages the major drawback comes in the form of the fact that as the power demand falls below 80% the fuel consumption of the gas turbine increases where as the amount of steam generated from WHRB drop significantly. To overcome the decreased supply of the steam form such system generally a supplementary firing system is maintained in such WHRB systems. The burner for such boiler are provided in the exhaust flue duct which link the gas turbine with WHRB. This ensures the supply of heat in the same amount even if the gas turbine is not fully loaded. The gas turbine configurations are being explored widely by integrating various cogeneration systems apart from simple steam generating units and various possibilities are being explored to improve the existing system. This provide large and prominent field of research to be carried.

1.4.3 Reciprocating base cogeneration system

They work on the principle of IC engine and is quite similar to gas turbine systems here also the reciprocating engine fired through fuel to run a generator. The generator produces electricity while the engine releases the high temperature exhaust gasses which can be utilised for production of process steam. Apart from exhaust gasses, the cooling water heat exchanger as well as lubricating oil in this case also acts as a heat supplier from which heat can be recaptured to generate process heat. These system are available with variable speed system i.e., the system can be low speed engine type can be medium speed engine type or can be high speed engine type. Following Figure 1.4 shows one such typical reciprocating engine based type cogeneration system. Although the low speed reciprocating engines comes with higher efficiency in compared to the medium and high speed engine but the temperature of exhaust as well as the quantity of the exhaust is relatively less as a result for cogeneration systems, medium and large scale reciprocating engine are preferred as they provide flue gasses with higher temperature as well as in large quantity.

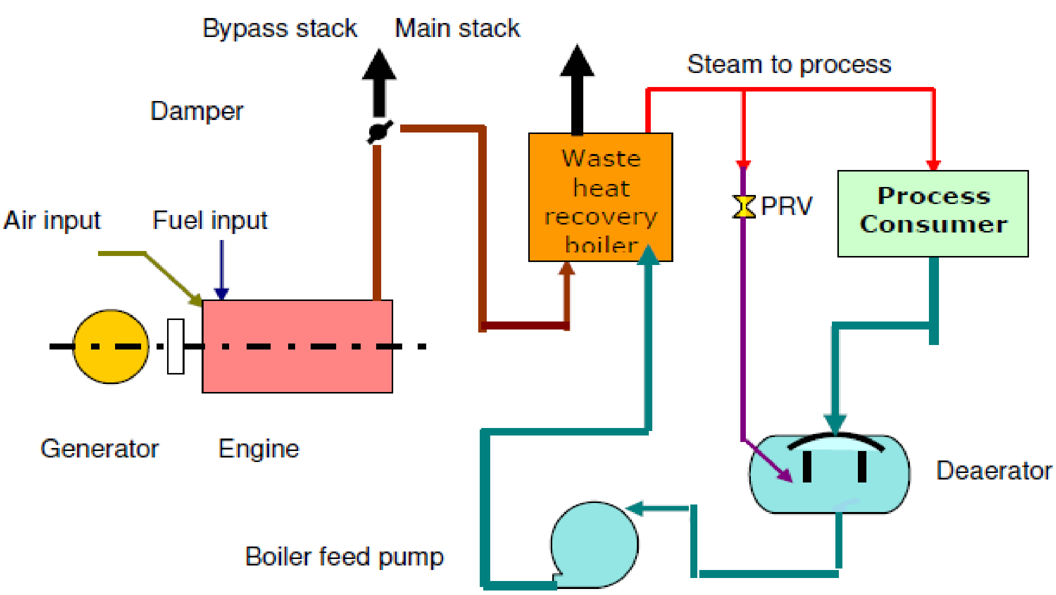


Figure 1.4: Reciprocating engine based cogeneration system (Reference [2])

Generally when electricity is generated using diesel engines a substantial amount of heat is lost from the system. Integration of cogeneration system helps in recovery of the heat from lubricant cooling system, water jacket cooling system in addition to heat recovery from the exhaust through manifestation of process steam or heated water. With integration of cogeneration system, the efficiency of the system is increased from 35%-42% to 65%-75%. These system are considered highly suitable when there is high requirement of electric power in compared to steam i.e., systems requiring high power to steam ratio, as reciprocating engine based cogeneration system produced more heat rate per cycle as compared to steam as well as gas turbine cogeneration system. This reciprocating engine can only be fired using hydrocarbon fuel only mostly high speed diesel, light diesel oil, residual fuel oil and natural gas are commonly used. Use of natural gas has recently gained the traction, for being used in reciprocating engine, as it is comparatively a clean fuel and requires no engine modification. However the system give lower efficiency in compared to steam based as well as gas based cogeneration system as a result it is not considered economically suitable especially when there is continuous demand of power and steam. In addition to that the reciprocating engine demand high maintenance as a result it requires periodic stops which makes them suitable for cases where there is discontinuous requirement or as a stand by source only for occasional use.

1.4.4 Combined steam and gas turbine based Cogeneration system

It has two fundamental principle integrated in one system i.e., Rankine cycle and Brayton cycle integrated together. Figure 1.5 shows a typical combined cycle based cogeneration system.

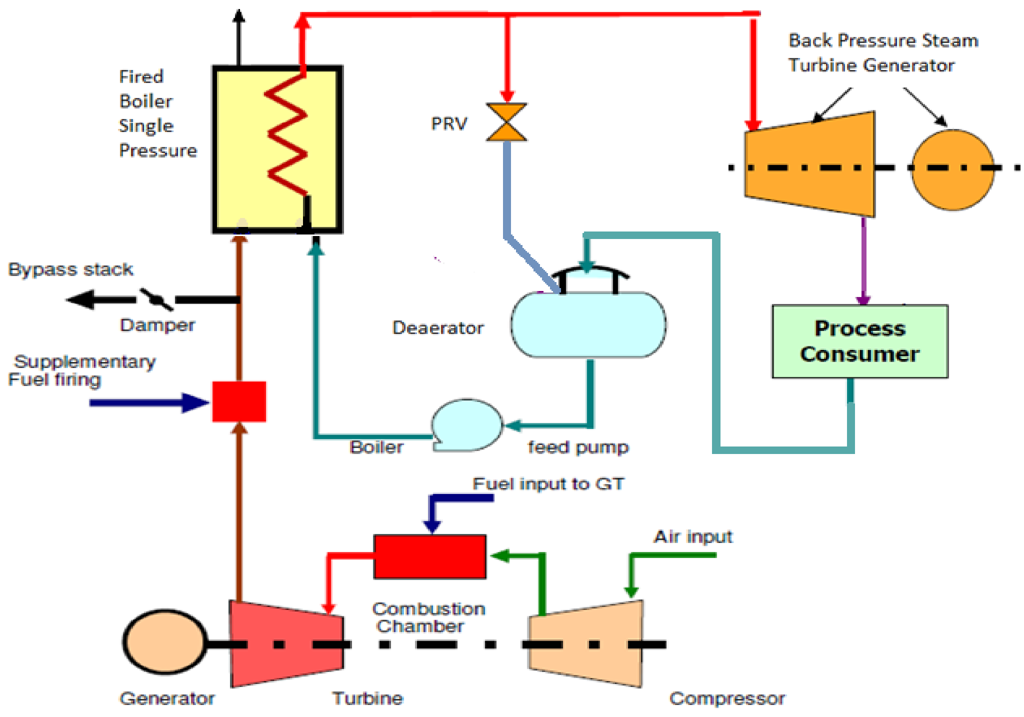


Figure 1.5 : Combined steam and gas turbine based Cogeneration system (Source : Reference [2])

Here the gas turbine is run by burning fuel and the exhaust gasses coming out of the gas turbine are then used for generating steam through utilisation of WHRB systems which generates high pressure steam. This steam with high pressure and temperature can be used in three different manners to generated electricity. It can be though a back pressure steam turbine, through back pressure steam turbine with extraction unit or through an condensing steam turbine with extraction unit. Steam obtained either through extraction or in the form of low pressure exhaust steam is supplied to the process consumer for utilisation. This system comes with very high furl utilisation factor in compared to other cycle and hence is one of the most promising field explored for practical applications. Combined cycle power plants provide better performance in compared to back pressure steam turbine system when ratio of electricity generation to thermal loading is very high as extra power is supplied by the back pressure system turbine system while the main power supply is maintained by the gas turbine, without using extra fuel supply. The only drawback of the system is same as that of gas turbine based cogeneration system that is less operational flexibility when the load on the gas turbine is lowered. To reach the flexibility of extraction cum back pressure steam turbine supplementary firing system can be placed to keep the supply of steam maintained. In this type of cogeneration system the condensing steam turbine with extraction unit finds its application in areas where persistent requirement of electricity continues to be very high where as the requirement of steam remains limited. The amount of process steam obtained in through the system is controlled by using bypass valve, so when there is no requirement of steam the attainable value of electrical power output can be enhanced by allowing the complete exhaust steam to flow into the condenser for that period of time. When process steam is required at different temperature and pressure condition then such conditions are met by installation of condensing steam turbine with more than one extraction unit mechanism or by using the configuration of back pressure turbine with one or more extraction units. Through this cycle an overall plant efficiency of 90% can be achieved with optional fuel utilisation. The lower heat rate along with economical specific capital cost of gas turbine and accessibly to the power generated from open cycle use of gas turbine power plant makes combined cycle plants highly economical. Only drawback comes in the fact that it has very less fuel flexibility.

1.5 TECHNO-ECONOMIC ADVANTAGES OF COGENERATION SYSTEM

Following are the major techno-economic advantage of utilisation of cogeneration system in the industries to meet the energy requirement:

1.5.1 Cogeneration system promotes the very notion of energy conservation through highly efficient utilisation of limited fuel source through optimisation study which is carried out before installation of project in any industry.

1.5.2 The total cost associated with steam and power output become comparatively small as cogeneration system favour lesser capital cost and low operating cost as well as it decreases the recurring cost which makes the system attractive for the industries.

1.5.3 These industrial cogeneration plants supplements the power generating units of the state and thereby help in fulfilling the gap between supply and demand by in house power generation.

1.5.4 The utilisation of cogeneration system in various industries can also help in reducing the transmission and distribution losses as they can provide the electricity supply at the vicinity.

1.5.5 Cogeneration systems in an industry makes them self reliant by reducing their dependencies on state power generating units and irregular fluctuation that occurs due to improper supply of electricity from state board.

1.5.6 The cogeneration system promotes efficient utilisation of fuel and thereby reduces the environmental pollution as the overall fuel consumption is reduced; this makes this system very environment friendly and is in line with the current needs.

1.5.7 The variability of fuel that can be burned to produce steam in steam turbine based cogeneration system promotes the use of alternative sources of fuel such is rice husk, sugar bagasse etc to be used as fuel which can help in conserving the available natural resources.

CHAPTER 2

GAS TURBINE COGENERATION SYSTEM

2.1 INTRODUCTION

Gas turbine prospered in our country in last two decades only, this makes the system relatively new in the industries, especially in India. The main reason for choice of a gas turbine based cogeneration system is the ease in the starting and shutting down of the system in addition to it the system requires less maintenance apart from it the gas turbine cogeneration system provides with significant proportion of heat in the form of exhaust which can be employed straight into heating application or indirectly through steam generation as well. When there is demand of energy in the industries in the form of direct heat or in the form of steam along with requirement of electricity in such a manner that the power to heat ratio is smaller than 1 then gas turbine cogeneration system are considered an optimal solution for optimising the cogeneration system being used for energy conservation. Based on the rating of the gas turbine, the efficiency of the gas turbine system can range between 24-30% [2]. As a result a plenty of heat energy is wasted through the flue gasses from the gas turbine generator. If this energy is tapped through proper technical means, it can lead to increasing in the global performance of the system up to 85% [2]. Although gas turbine has its perks still it is not considered suitable for application in system where there is considerable fluctuation in load, at a load of less than 80%, gas turbines don't perform efficiently, the fuel consumption goes on increasing as well as the amount of heat recovered from secondary system also goes on decreasing. To obtain the best performance form gas turbine based cogeneration system the availability of constant load and constant demand of process heat are favourable conditions. The general configuration of such plant includes gas turbine with its exhaust attached to a WHRB. The WHRB can be of different type that is unfired, supplementary fired or fully fired. Recently several new systems have been explored to advance the gas turbine based cogeneration system for e.g., absorption chiller based cogeneration system used for generating cooling load through waste heat recovery.

Table 2.1: Various gas turbine and waste heat recovery systems used for cogeneration (Source: Ref. [2])

Generator in Gas Turbine	**Method of Heat recovery**
Natural gas based plant	Steam generation in unfired/supplementary fired/fully fired WHRB
Liquid fuel based plant	Application of steam straight in process
	Application of steam for power generation through steam turbine generator [Cogeneration-cum-combined cycle]
	Absorption Chiller [Cooling heating and power System]
	Application of heat for direct space heating

Table 1 shows all the major system of heat recovery used in gas turbine cogeneration system. The generator of the gas turbine generally uses natural gas, high speed diesel, light diesel oil, naphtha, etc as fuel source, other fuel sources such as furnace oil can also be used but the systems performance get effected, i.e., the gas turbine based cogeneration system comes with lesser fuel flexibility.

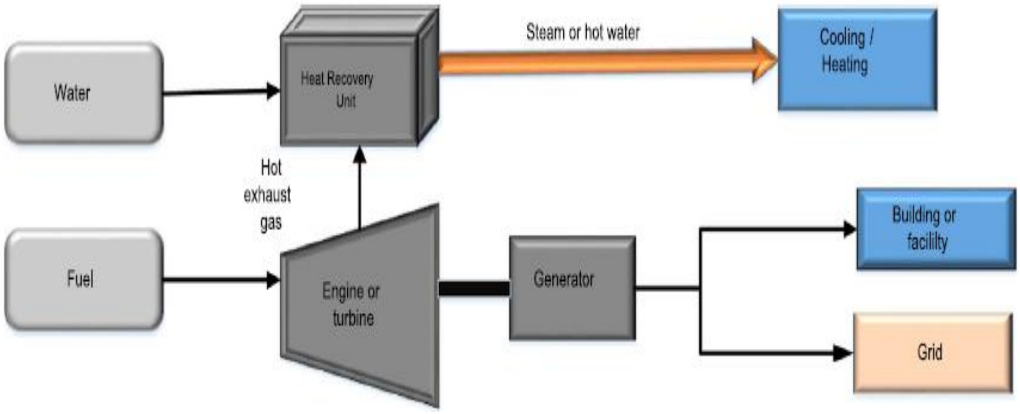
2.2 CONFIGURATION OF GAS TURBINE BASED COGENERATION SYSTEM

There are two fundamental configurations of gas turbine based cogeneration systems :

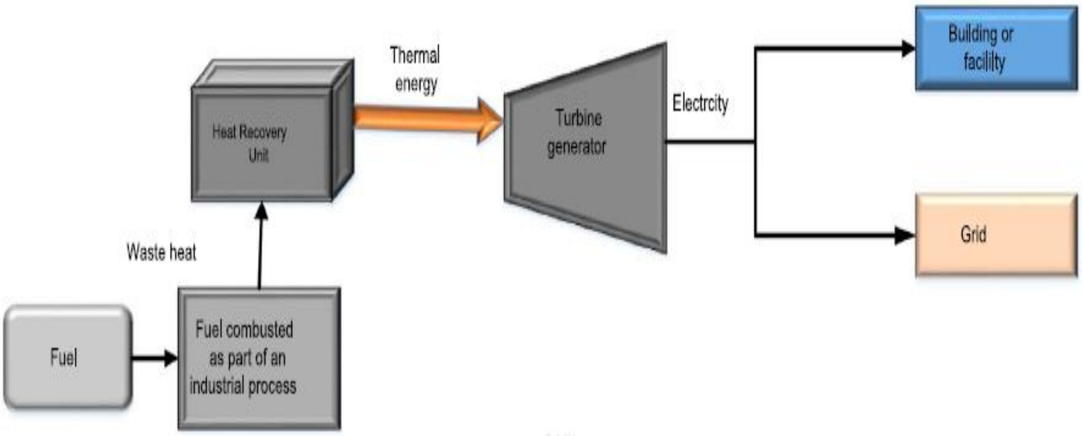
1. Topping-cycle
2. Bottoming cycle

2.2.1 Topping-cycle

In this configuration, the fuel is employed to generate electricity or mechanical power in the prime mover. After the electricity or mechanical power is obtained, the exhaust so obtained is used for providing the process heat, generating steam through WHRB or for direct heating of a given space. The electricity generated here can be used by the facility itself or can be supplied to the grid. The main purpose of topping-cycle gas turbine power plant is the generation of electricity for on-site application such as on chemical or metal industries and the exhaust obtained through generator is used for generation of process heat, electricity generated in this configuration can be utilised for on-site application or can be sold to the electricity grid. The electricity generation is secondary in nature for such plants. Such plants generally demand some additional fuel supply to generate the electricity which has to be included in the operational cost of the cogeneration system.



2.1 (a): Topping cycle



2.1(b): Bottoming cycle

Figure 2.1: Topping and Bottoming cycle of gas turbine cogeneration system (Source: Reference [4])

2.2.2 Bottoming Cycle

Here the prime purpose is the generation of heat for manufacturing process in any industry and can be carried out through any fuel or any other heat producing chemical reaction and the exhaust recovered from such process is then used for electricity generation. This kind of bottoming cycle is also known as waste to power generating cycle and is usually used in industries such as steel industries or glass industries where high temperature of flue gasses is required to carry out the manufacturing process and a portion of exhaust heat is recovered through power generating unit which can supply the electricity to the facility or it can distribute it through the grid. Figure 2.1(a) and 2.1(b) represents the topping and bottoming cycle configuration of gas turbine based cogeneration system.

2.3 GAS TURBINE COGENERATION SYSTEM UNDER STUDY

Following are a few of the system which has been under continuous investigation for research in field of gas turbine based cogeneration system.

2.3.1 SOFC-GT cogeneration system

It was Ide et.al., [10] who introduced the SOFC-GT based cogeneration system. In his work investigation was carried out to examine the plant efficiency and the power generation through plant in case of three fuel cells. The fuel used for analysis was natural gas; it was found that the major loss of efficiency of the system occurs due to presence of highly irreversible combustion of fuel. With this experiment SOFC-GT cogeneration system saw significant advancement, it was in 1990 when for the first time large scale SOFC based power generation units were established. Soon the research sifted from operating the SOFC at atmospheric conditions to operating it at pressurised condition. Harvey S.P. et.al., [11] for the first time started to analyse a pressurised SOFC model using ASPEN plus simulator, in this work they tried increase the efficiency by reducing the overall exergy losses through the system, this research was carried out at Argonne National Laboratory operated by the University of Chicago, while in 1992 they used exhaust gasses as oxygen carrier to reduce the exergy losses in 1994 the used a recycled exhaust gasses as a carrier of oxygen to reduce exergy losses and showed how the efficiency is increased in later case. However it was the Siemens Westinghouse, a leading manufacturer of pressurised SOFC which manufactured the first prototype of pressurised SOFC-GT cogeneration system in 1997. The system was capable of producing 220KW of power. Since then several analysis and research has been carried out for thermodynamic and economic optimisation SOFC-GT based cogeneration system to increase the feasibility of the system. Various parameters effect the layout of the system it mainly includes pressure of operation, temperature of operation, internal and external reforming type, and type of gas cycle combined with it. There can be four type of basic configuration of SOFC-GT based cogeneration system depending upon position of gas turbine it can topping or bottoming cycle and based of operating pressure t can atmospheric or pressurised cycle.

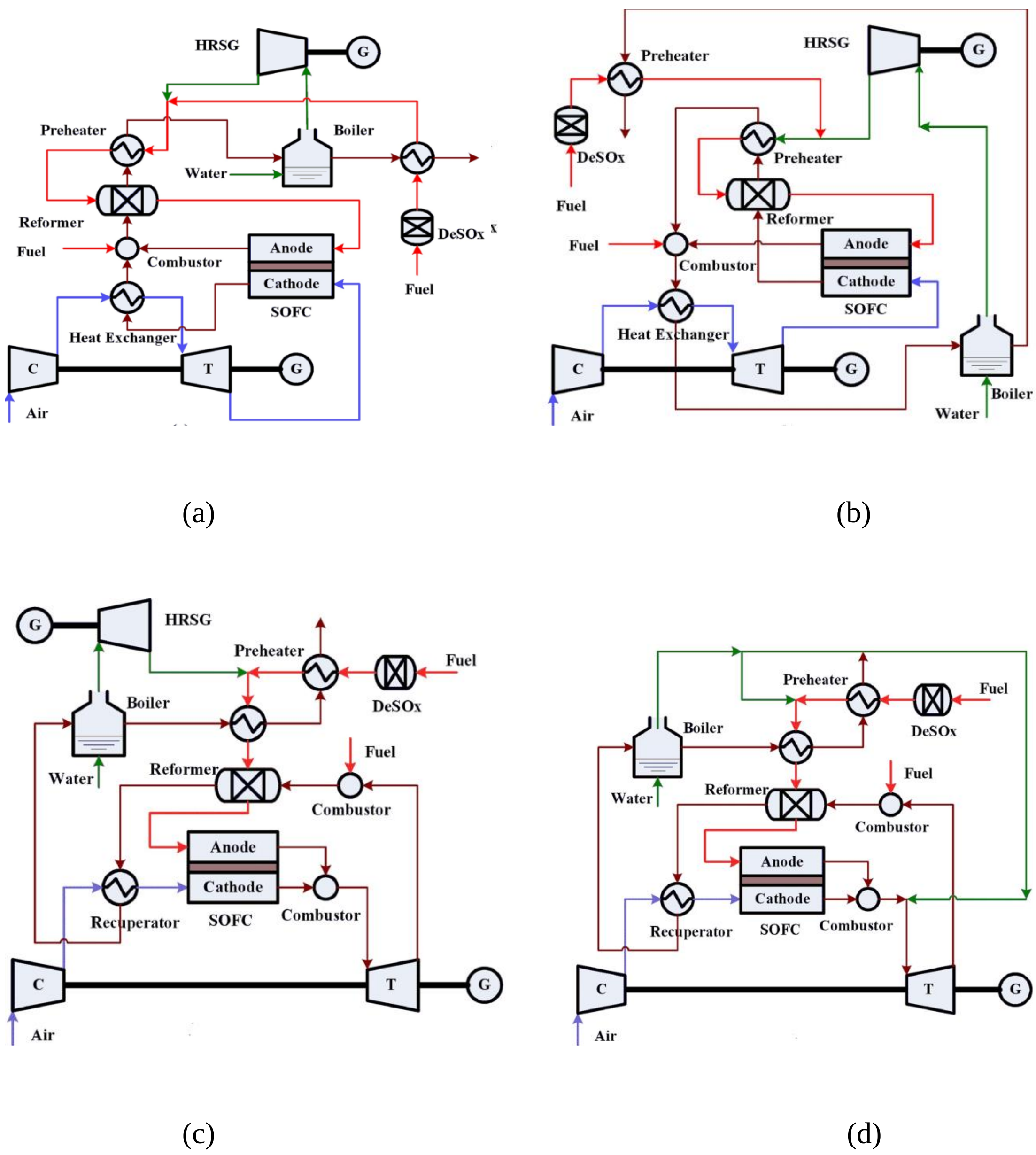


Figure 2.2: Four basic layouts of SOFC-GT cogeneration system. (a) & (b) pressurised, (c) & (d) are atmospheric, (Source: A. F. Massardo and F. Lubelli [12])

Figure 2.2 depict the four common configuration of SOFC-GT cogeneration system. The two main component of the system, i.e. SOFC and GT can be assembled in two different manner, in case of bottoming cycle the exhaust of gas turbine is manifested as the heat source for the SOFC system where as in topping cycle the SOFC itself act as a combustion chamber for GT system. The main feature of bottoming cycle lies n the fact that the two units are independent of each other it makes integration and disintegration of the system easier and makes one system immune to anomalies that occur in the other. In contrast to that in the topping cycle the two system as integrated as one unit making one system dependent on other. Through comparison of the two system it has been concluded that topping cycle has higher efficiency however it comes with the challenge of operating SOFC system at high pressure which makes the choice of material for SOFC difficult and makes the system costly. Figure 2.2(c), adds the system of heat recovery steam generation system to the cycle, which integrates the Brayton cycle with the Rankine cycle. Figure 2.2(d), represents Cheng cycle, which introduces steam as an additional working substance to the gas turbine to improve the working of high-temperature gas. Top cycle has remained more popular in practical applications of SOFC-GT system models..

2.3.2 Gas Turbine - Combined Cooling Heating and Power Cogeneration System (GT=CCHP)

Although the CCHP system has been in use with several methods of hear input such as solar heater, ICE etc the utilisation of gas turbine exhaust as the heat source for CCHP system came into picture quite recently. This makes this area new and is being explored repeatedly as an alternative to conventional domestic or commercial cooling system. Kong X.O. et al. 2005 [13] in his work presented a simple integration of CCHP with gas turbine system, he investigated the performance of the system through parametric analysis, from this work it was deduced that better test ring platforms can be obtained in case of small scale CCHP plant instead of large scale plants. This promotes to carry out research for management and optimisation of such system. The system allows to vary the cooling load, heat load and electricity load easily in accordance with the requirement. This laid the foundation of further research in this area. In his next work, Kong X.O. et al. 2005 [14], economic optimisation of the system was done along the line of energy management of the system. In this work, to determine optimal energy management strategy a linear model for the complete cycle was suggested and the overall cycle cost energy function was minimised. Form the analysis it was concluded for effect cost management it is advisable not run the gas turbine in certain conditions. The model prepared by them was unable to predict the thermodynamic properties of substance at various point in the system due to which exergy analysis of the system was not carried out in this model. Yeng et,al. [15], in his work present the CCHP system using gas turbine as power generator, HRSG as heat generator and absorption chiller to incorporate the cooling load, in this work thermodynamic analysis and economic analysis of system was carried out , system's first law efficiency, second law efficiency and economic cost was optimised. The paper concluded that higher economic thermal and energy efficiency can be obtained if the system is operated at higher power output and at higher cooling load. Liu et al. [16], in a similar work presented a comparative performance, by investigating the hourly analysis of CCHP system integrated with GT or ICE as the heat source. On the basis of hourly analysis it was deduced that the CCHP system are dependent on the transient environmental conditions and the optimum operating strategy should be designed keeping in mid the surrounding condition and its effect on the system. Paper proposes to install a thermal storage system in place of heat or steam generating system for heat recovery as it reduces the operation cost of the heating system. A similar work carried out by Ebrahimi et.al. [17], consisted of CCHP system integrated with a industrial gas turbine, the work involved the energy, exergy and environmental analysis of the system. The places of constant load requirement are considered suitable for the application of such GT-CCHP based cogeneration power plants. Li et.al. [18] , in his work has carried out analysis of CCHP system integrated with GT and ICE for different places of constant load including hotels, offices and residential building. In the comparative study between different location it was established through the result that the system perform batter in case of hotels where fluctuation in the load on the system is comparatively less. A general configuration of the system discussed has been depicted in figure 2.3. The given system represents a CCHP integrated with a gas turbine which is coupled with a double effect absorption chiller and water heat exchanger. The fresh air at point 1 is first mixed with flue gases coming out at 9, this mixture are then supplied to the air compressor (COMP). Compressor gives the air at high pressure and temperature at point 3. This compressed high pressure air is then supplied to combustion chamber (COMB), after addition of the fuel burning takes place as a result of which exhaust gases are formed. These combustion products are represented by point 4 in the diagram. These gases are then supplied to the gas turbine (TUR). As these gases are expanded in the gas turbine to lower pressure, power is produced. The exhaust gasses obtained at the outlet of gas turbine is then supplied to the absorption chiller (ABC) and water heat exchanger (HX). A portion of exhaust is taken out at point 9 and the fresh air entering at point 1 are mixed together. In the given system Lithium Bromide and water absorption system is used with double effect absorption chiller.

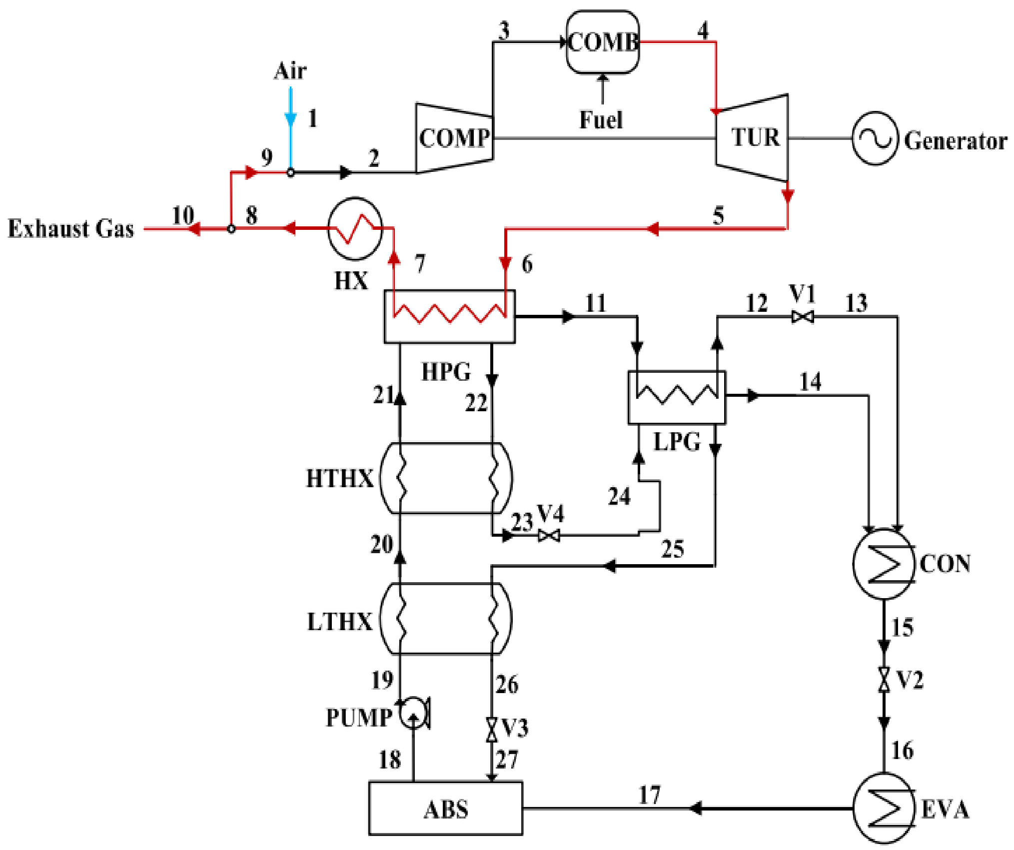


Figure 2.3 : Schematic diagram of CCHP-GT cogeneration system (Reference [19])

Thermodynamic modelling of the given system is described in reference [19]. In lithium bromide and water system first regenerated dilute solution at 18 is heated, this preheating takes place in two steps, first heating is done in low temperature heat exchanger(LTHE) then at high temperature heat exchanger (HTHX) after which the solution is fed into high pressure generator (HPG). From the high pressure generator, high pressure refrigerant vapour at 11 and medium concentrated solution at 22 get unconnected from the low concentration solution which enters the high pressure generator at 21. This medium concentrated solution is then supplied to the low pressure generator (LPG) from which again high concentration solution is extracted. The refrigerant extracted at 12 and 14 are the supplied to the condenser (CON), where they are allowed to convert from gaseous state to liquid state. This liquid refrigerant is then throttled by an expansion valve (V2). As the refrigerant expands from higher pressure to lower pressure its temperature decreases, this refrigerant then goes into the evaporator (EVA) at which the cooling effect is achieved.

2.3.3 Gas Turbine – Organic Rankine Cycle (GT-ORC) cogeneration system

Organic Rankine cycle (ORC) is conceptualised on the same working concept as that of steam Rankine cycle, the only difference lies is in the type of working fluid. On one hand where water finds its application as working fluid in conventional Rankine cycle, ORC fins the application of organic fluid for power generation. The main difference lies in the low heat of vaporisation of the organic fluids which make them excellent medium for heat extraction form low temperature sources i.e. from heat source of temperature less than 230 ℃. This system has evolved over the time as many researches have been carried out for optimisation of the system along with selection of desirable organic fluid for the system. Generally, a dry isentropic organic fluid is considered suitable for the system as superheated state of such fluids are not required to be achieved for obtaining the desired efficiency. The earlier research in this field was concentrated remained very restricted as the economic feasibility as well safety of use of organic fluid in the systems was major concerns. Hung T.C. et.al. [20], in his review has highlighted few of the important work carried out in early phase of development of ORCs as a waste heat recovery units. The paper has shown the analysis of different working fluid and has detailed the perks of using dry organic fluid over the wet organic fluid. The fact that saturation curve of dry working fluid remains almost vertical leads to almost negligible condensation while expansion process takes place through turbine and makes them ideal fluid for ORC system. Hung in his work has also carried out parametric analysis of wet and dry organic fluids and concluded about the favourable range for which the wet working fluid will be more suitable in compared to dry working fluid and vice versa. However, the paper also points out the issue related to stability and safety of using organic fluid at high temperature and pressure condition. As a result of such concern the system was unable to see major breakthrough for a considerable duration of time. It was in 2001 when for the first time Yamamoto T. et.al. [21], in his work gave numerical simulation model of the ORC along with the experimental model of ORC. In his work he made a comparative study of HCFC-123 and water as the working fluid foe recovery of low grade heat energy. The result obtained in the paper established the fact of betterment of organic fluid over water as a source for recovery heat energy lower temperature sources. It work also concluded that the best operating condition for any of the fluid with low latent heat of vaporisation would be having saturated vapour at the turbine inlet condition. Along with the parametric analysis several other work related to selection and testing of various organic fluid for the ORC system started to gain traction. First of such work was performed by Maizza et.al. [22] , in this work the several organic fluids were examined for their effect on overall system efficiency and the plot was developed to select the desirable fluid under variable condition. It was in 2001 only when the research work on ORC system integrated to gas turbine as its bottoming cycle started to emerge. Najjar Y [23], presented few of those starting work. In his paper he analysed and studied the data and results of various gas turbine based cogeneration system including CCHP, steam based rakine cycle as well as ORC as the bottoming cycle. He tried to study the combined cycle with two different configurations which included single shaft and double shaft gas turbine. While using ORC as bottoming cycle he studied four organic fluid including R12, R22, R113 and R114. In his work the parametric study employed to note the variation of total work output in respect of variation in compression ratio of gas turbine, gas turbine inlet temperature and pressure ratio of bottoming cycle. The paper concluded that the combined cycle plant has the ability to augment the overall efficiency near to 60%. It also resulted into superior power output and better economics. In case of ORC based combined cycle R22 was observed to be performing better than other organic fluids. Although there were several works carried in field of ORCs no significant work was seen in integrated system till 2008. After 2008 onwards the work in the field of integrated gas turbine and ORC based cogeneration system gained its traction. Figure 2.4 shows a schematic representation of gas turbine cogeneration system integrated with ORC as the bottoming cycle. The system has same configuration as a steam based combined cycle. The ait through air through air compressor is first fed into the pre-heater. The heated air is then delivered to combustion chamber and the fuel is supplied and then the air with high temperature and pressure is supplied to the turbine. After expansion through the turbine the exhaust gasses are then fed into the heat exchanger at which the gasses exchange heat along with the organic fluid. Generally the mass flow rate is maintained in such a manner that the output obtained from the heat exchanger is saturated organic fluid vapour.

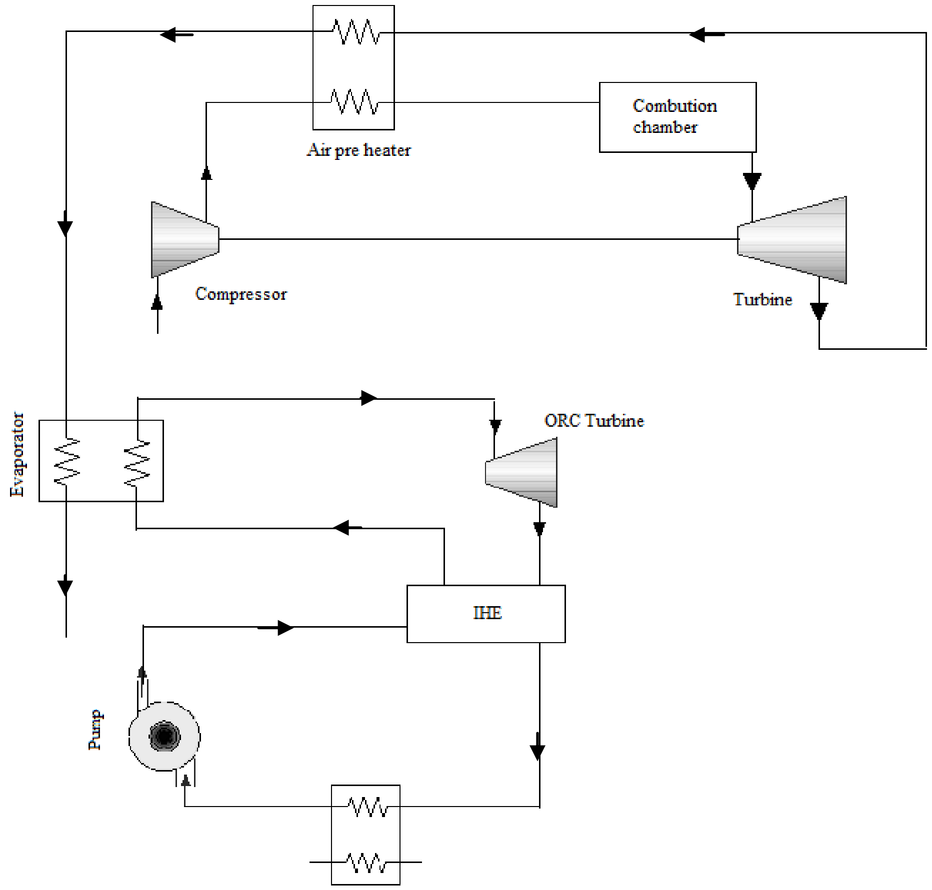


Figure 2.4 : Gas turbine cogeneration system with ORC as bottoming cycle (Source: Reference [24])

This vapour is then fed into the ORC turbine where it is expanded to lower pressure. The expanded fluid is then fed into the condenser from where the condensate is extracted and liquid organic fluid is then compressed to higher pressure again through pump.

CHAPTER 3

LITERATURE REVIEW

Following are the major literatures which were reviews for development of the proposed cycle in the project:

Mortaza Yari (2008) [24], has worked on the combined cycle of micro gas turbine integrated with micro ORC system. The ORC system has an internal heat exchanger (IHE) attached to it. In his work parametric analysis of the system was performed with variation in parameters including turbine inlet temperature of turbine, compressor pressure ratio of system, the evaporator temperature difference and variation in degree of superheat provided to the ORC system. The organic fluids used in the system included Isopentane, n-heptane, n-octane, n-hexane, R123, R113 and toluene. He plotted the variation of efficiency with respect to overall power generation of the system at different values of parameter under study. The conclusions drawn from the work included, enhancement in turbine inlet temperature increases the overall efficiency however the work output decreases as the amount of work recovered by bottoming cycle goes on decreasing. The increase in the inlet temperature of ORC turbine increases both the work output along with the performance of the system, however as the difference in the temperature of the heat source and the organic fluid increases in the evaporator both efficiency and work output suffers. Superheating also creates an adverse impact on both work delivery of the system and efficiency. The work also compared the efficiency of the combined cycle and combined cycle having ORC with IHE. The results shows that combined cycle with IHE has efficiency O.5%-point higher that combined cycle with ORC not having IHE. Among comparison between different working fluid highest work output was obtained in case of toluene where as the iso-pentane was the worst performer.

Chacartegui R. et al.(2009) [25] , for the first time presented the integration of ORC with medium and large scale gas turbine with recuperative system, these system are capable of generating exhaust at a lowered temperature than conventional gas turbine. He took isobutene, cyclohexane, R113, R245, toluene and isopentane as the working fluids for ORC system for analysis. The system had the ORC with IHE and the configuration of gas turbine was also varied by introducing two stage compressions process and reheat in second part of the analysis. The paper studied the dissimilitude of combined cycle efficiency along with the change in inlet temperature of turbine for different configuration and with use of different organic fluids. Among the various fluid toluene showed the best performance. It was deduced that the desired global efficiency of the cycle can be achieved at considerably lower turbine inlet temperature if ORC is used in bottoming cycle instead of conventional steam turbine based system. This conclusion can have noteworthy change in the designing of the gas turbine system making its manufacturing cost comparatively lower. The two configuration studied in the paper had shown that, the system with gas turbine having single stage compression and no reheat with ORC as bottoming cycle perform 4%-point better than conventional cogeneration plant with steam turbine as bottoming cycle and configuration having two stage compression and reheat system with ORC as bottoming cycle performs 3%-point better than conventional cogeneration plant with steam turbine in the bottoming cycle. The economic examination of the system concluded that the estimated increase in the cost of the system due to use of ORC as bottoming cycle will be compensated through increased efficiency of the cogeneration system. The major challenge highlighted in the paper was the utilisation of suitable ORC fluid in the plants as they are flammable in nature can be a risk to the safety of the plant and the working personnel.

Munoz de Escalona et.al (2012) [26], In his work the performance of cogeneration power plant with gas turbine in topping cycle and ORC in bottoming cycle was studied at part load conditions. The analysis was done by using 5 commercial gas turbine which included General Electrical LMS 100PA, Rolls Royals TRENT 60 DLN, solar CENTAUR 50, Solar MERCURY 50, Turbomeca MAKILA. For the analysis a mixed control strategy was taken into consideration i.e., an optimum operating was tried to be achieved through maintaining the efficiency of ORC along with maintaining the efficiency of heat recovery vapour generator as the variation of one was having a negative impact on other. For study multiple organic fluid were analysed to compare their performance in bottoming cycle and it was obtained that toluene perform batter in compared to other fluids. So toluene was employed as the working fluid for rest of the analysis in part load conditions. The paper concluded that TRENT, MAKILA, LMS 100PA showed no significant variation in overall efficiency even with installation of ORC as the part load condition were varied and it remained almost constant. CENTAUR 50 and MERCURY 50 rather showed significant variation they performed better at low load condition however as the load went on increasing the performance of the two turbine deteriorated. The significant observation was the inverse relation between the gas turbine efficiency and power output of ORC as the turbine efficiency decreases the heat available for recovery at ORC cycle increases so, it act as a noteworthy contributor in maintaining the global efficiency of the power plant.

Ahamadi et.al.(2012) [27], in his work Ahamadi proposed a tri-generation system consisting of heating cooling and electricity production simultaneously. The system consisted of a gas turbine combined with ORC for electricity generation, single effect absorption chiller for meeting the cooling load and domestic heater for meeting the demand of process heat. The parametric along with the environmental examination of the system was performed to understand the behaviour of tri-generation system. Under parametric study the authors did both energy along with exergy analysis, under which the change in system behaviour was studied by varying the inlet temperature of turbine, its compression ratio of system, the isentropic efficiency of turbine and turbine inlet temperature of ORC. The result generated in the study depicts that an optimum value of compression ratio is achieved during analysis as the it balance the overall efficiency by two contrasting parameters i.e., decreased fuel supply and increased work consumption by the compressor. The inlet temperature of gas turbine has significant effect on the system as increment in the inlet temperature of gas turbine increases both first law efficiency and exergy efficiency. However with increment in turbine inlet temperature the exergy destruction as well as fuel consumption also increases which creates a negative impact on the system. The environmental analysis of the system shows an reduction in CO_2 emission in compared to the cogeneration system consisting of gas turbine and CCHP integrated together.

Stefano Clemente et.al. (2013) [28], In this work the detailed definition of the system design and main feature were provided for integration of the ORC to a gas turbine of 100kWe integrated with internal recuperator. In this work the system was optimised for six different working fluids. The fluids considered for the analysis included R254fa, siloxanes including MDM, D4 and MM along with Isopentane and Isobutane.

Following the optimisation part the work was further carried out to find out that how system varies with change in the type of expander used in the system for power generation. Single shaft configuration of both axial flow and radial flow turbine were analysed, with each designed taken into consideration the type of working fluid that will be used. This work also provides an insight into the method of design of various expanders that can be used in the ORC system. It was established in the results that isopentane was the best performing organic fluid irrespective of the configuration used. The installation of ORC in the bottoming cycle helps in improving the efficiency by 8%. The paper highlights the choice of the configuration of the turbine should be based on the smooth integration of system with other equipments mounted in the shaft. The paper also concluded that siloxanes creates problems while being operated at high rotational speed and hence are suitable only when rotational speed of the ORC turbine is below 3000rpm.

Carlo Carcasci et.al.(2014) [29], this paper provides a simulated examination of ORC acting as a bottoming cyele to gas turbine system. In this work four working fluid were analysed which included toluene, benzene, cylcopentane and cyclohexane. For safety reason the authors used a for heat transfer a diathermic oil circuit between the two plants in the cogeneration system. Work was carried out to perform the comparative study between the four working fluids to recognise the best choice for the system under consideration. The authors tried to optimise the cogeneration system for different operating pressure of working fluid along with different oil circuit temperature. In the system the use of superheating was also investigated. It was observed that for toluene, cyclohexane and benzene the system performs better when at the entry of the turbine the fluid is in saturated state but in case of cyclopentane the super heater installation improves the efficiency of the system. The increase in the temperature of oil circuit also have a favourable effect, it increases the total power output of system. This increase is only restricted by the heat recovery oil capacity limit. It was also concluded that the oil temperature range determines the best performer among the various working fluid studied. At lower oil temperature while cycloheaxane is the best performer, at higher oil temperature toluene is preferable.

Le et.al., (2014) [30], In the paper two system configuration one of basic supercritical ORC and other of regenerative type supercritical ORC was used to study different organic fluid with less global warming potential which included R134a, R152a, R32, R744, R1270, R290, R1234yf and R1234ze(E). The comparative study between the two system was done by analysing each fluid for both basic supercritical ORC as well as for regenerative type supercritical ORC. The paper provided first for the basic model of supercritical ORG without regeneration system. It also provides for the algorithm used in calculating the mass flow rate of the fluid through iterative method in case of supercritical ORC. After modelling the basic supercritical ORC it was integrated with regenerative system. The relative study between the two systems and comparative study between various organic fluid shows that, maximum optimal efficiency is obtained in case of R152a in both the cases of regenerative as well as basic supercritical ORC. However, highest value of optimal power was obtained with R1234ze. The comparative study between the two configurations establishes that the regenerative type supercritical ORC performs better than the basic supercritical ORC.

Khaljani et.al. (2015) [31], in his work carried out an multidimensional optimisation of the system to obtain the desired design parameter. The exergy efficiency as well as complete cost rate of the system were taken as the primal functions to carry out the analysis and non dominating shorting genetic algorithm – II was utilised in the analysis. The remaining parameter such as inlet temperature at turbine, compression ratio of gas turbine, APH outlet temperature, efficiency of gas turbine, compressor efficiency, pitch point temperature difference of HRSG and evaporative system, etc were considered as decision variables for the analysis. The work analyses four working fluids which included R113, R123, R254fa and R600 to be used as the working fluid in the analysis and R123 was chosen because of its performance for exergy efficiency as well as less global warming potential. The results show that by using multidimensional optimisation the system efficiency was enhanced to 56.15% from 51.4% and the cost of the system was also significantly reduced.

Khaljani et.al.(2015) [32], This was the second work presented by the authors giving the basic structure for analysis of GT-HRSG-ORC cycle through EES software. In this work detailed analysis was provided by the authors of individual component for energy, exergy, economic and environmental aspects. The three objective functions taken for the study and optimisation were first law efficiency, exergy efficiency and cost of overall system. The paper describes in details the modelling of the system for thermodynamic, economic as well as environment related examination of system. The parametric study of the system was carried out by varying turbine inlet temperature, compression ratio, APH outlet temperature, efficiency of gas turbine, compression efficiency, ORC turbine efficiency, pump efficiency, pitch point temperature of HRSG and evaporator, ORC turbine inlet temperature and condenser temperature.

Cao et.al. (2016) [33], this paper had an objective of optimisation of designing and providing thermodynamic study of the GT-ORC combined cycle. In this work gas turbine was integrated with ORC in bottoming cycle with two recuperators. Aromatics were employed as the working fluid for the ORC system. One of the recuperator was placed at the topping cycle where incoming air was extracting heat from exhaust gasses and the second recuperator was placed at the bottoming cycle where the ORC turbine exhaust was exchanging heat with condensed working fluid. Considering the temperature of exhaust (>523.15K) three competing fluids were analysed to be chosen for the study of the system, these included alkanes, aromatics and siloxane. Considering the other factors like auto ignition temperature of the fluids p-xylene, toluene and ethylbenzene were admitted as working fluid of the system. For carrying out simulation for thermodynamic QD128 Gas Turbine was taken in the topping cycle. It was deduced that in compared to conventional steam turbine system the use of ORC as bottoming cycle augment to the efficiency by 6.4%. It was also found that toluene perform better in compared to reaming two working fluids. After this four more gas turbine were analysed in the topping cycle which included GE-10, SGT-200, Mercury 50 and Taurus -70. However, on optimising the thermodynamic performance of the turbine with change in pressure of bottoming cycle, QD128 emerged as the gas turbine be the giving most promising performance among all the gas turbine used in the work.

Yagli et.al.(2016) [34], in this work a comparative study was provided between the two system using supercritical and subcritical ORC as the bottoming cycle. For this a combined heat and power engine fuelled by biogas was used. This work was carried out taking R245fa as the working medium for both the ORCs. The paper tries to provide a positive relationship between pressure and temperature increase with respect to other parameters such as total work output, pump total power consumption etc. In this work the two systems was studies mainly as a temperature and pressure dependent function only. While analysing the supercritical ORC certain pressure range of below critical pressure was also taken under consideration so that transition between the two cycles can be captured during the analysis. After the parametric study the comparative study was also carried out. It was provided by the work that the supercritical ORC deliver better performance both on the exergy as well as energy analysis of the two systems.

Algieri et.al.(2012) [35], This work also presented a comparative analysis between the subcritical and trans-critical ORC. The topping cycle for above system was formed by a biomass based power plant; for bottoming cycle the exhaust obtained from the plant act as a source of energy and heat. The working medium considered for the analysis included cyclohexane, decane and toluene. The condensing temperature for the system was kept constant at 100℃. The subcritical ORC was analysed for saturated as well as superheated condition. Further the installation of IHE was also taken under consideration for analysis. In study of subcritical ORC it was observed that increment in the evaporator temperature has a favourable effect on overall system performance. Further the installation of the IHE in the system provides a significant enhancement to the overall performance of the system. It was also concluded from the study the superheating when applied to the system having IHE, creates a positive effect on the performance unlike the previous study where the system without IHE advocated the use of saturated vapour at turbine inlet. The comparative study also shows that utilisation of supercritical condition aids to the system overall performance and installation of IHE can further enhance it. The comparative study between the fluids shows that when subcritical system with IHE and superheating is used cycloheaxane comes out to be performing better than the other Organic fluids.

Sarkar (2018) [36], in his work provides for designing of pitch point and determination technique for both subcritical and supercritical ORC. In his work the simultaneous prediction of pitch point temperature for both evaporator and condenser is carried out in manner such that the heat recovery of the system is maximised. The paper highlights that the optimum operating pressure of for both condenser and evaporator is contingent on the mass flow rate of the working medium. Paper tells about the three method of designing of the system for given pitch point temperature difference. For supercritical ORC the iterative method was used to determine the pitch point temperature location. The paper has explored the method of design which evolves around the provided source fluid flow rate along with its inlet and exit temperature. The study concludes that with increment of the mass flow rate both work output as well as the heat recovered through ORC first goes on increasing and the goes on decreasing i.e., it attains a certain optimum point which gives the best possible value of mass floe rate for the maximisation of heat recovery. It also concludes that with increase in the heat source temperature, the work output as well as the heat recovery of the system increases.

CHAPTER 4

THERMODYNAMIC MODELLING OF GT-HRSG-SUPERCRITICAL ORC COGENERATION SYSTEM

4.1 INTRODUCTION

In thermodynamic analysis of GT-HRSG-supercritical ORC cogeneration system, first the thermodynamic model of the two system i.e., first GT-HRSG-ORC cogeneration system and second the supercritical ORC system was prepared separately and individual models were validated from the work carried out in previous paper. The GT-HRSG-ORC system was validated separately from Khaljani et.al. (2015) [32] and supercritical ORC system was validated from Le et.al. (2014) [30]. Then the two system were integrated to examine the system through parametric study of GT-HRSG-supercritical ORC.

4.2 SYSTEM DESCRIPTION

Figure 4.1 depicts the system described by Khaljani et,al. (2015) [32] in his work. The system consist of gas turbine in topping cycle and ORC in bottoming cycle with HRSG system for process heat recovery. The compressor take ambient air as in put, the pressure of the air is considered to be at 1 bar and temperature is taken to be at 298.15 K. The compressed air is then passed to the pre-heater; the outlet temperature of the pre-heater is considered to be at 850 K. Then air is passed to compressor chamber where fuel is injected into the system. The pressure for fuel injection is kept at 12 bar. After the combustion the exhaust is fed into the turbine where turbine inlet temperature is taken to be at 1520 K. The work output of the gas turbine system is kept unchanged at 30MW considering the turbine is working under full load condition. The exhaust gasses, after expansion through turbine is fed into the air pre heater from where it is passed to the HRSG system. In HRSG system the inlet condition, outlet condition and the water flow pressure is kept constant. The temperature at which water enters the system is kept at 298.15 K and the pressure of the supply is kept at 35 bar.

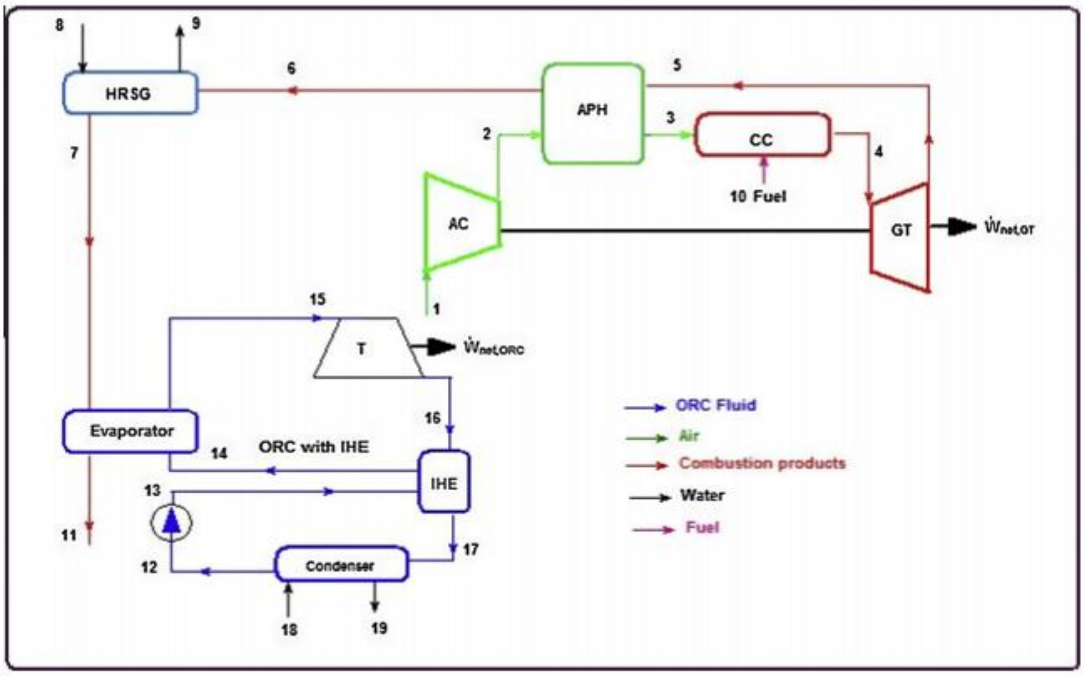


Figure 4.1: Schematic diagram of GT-HRSG/ORC system (Source: Reference [32])

After HRSG system the exhaust is then supplied to the evaporator of the bottoming ORC. ORC consist of 5 components including a turbine, IHE, condenser, pump and evaporator. The Turbine entry state of the organic fluid is kept at the saturation state of the fluid.

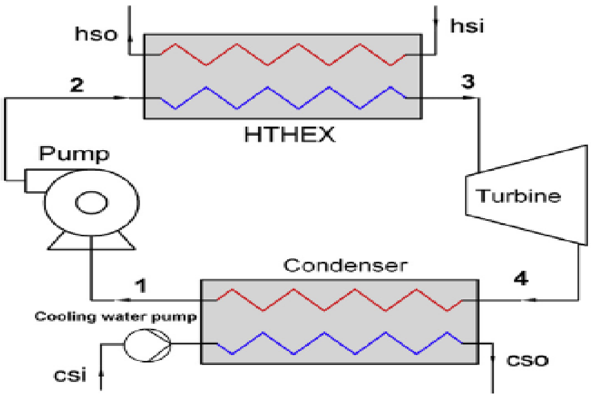


Figure 4.2: Schematic diagram of Supercritical ORC (Source: Reference [30])

Figure 4.2 represents supercritical ORC taken form Le et.al. (2014) [30], in the supercritical ORC the final pressure of the working fluid after compression process is higher than the critical pressure. The turbine inlet temperature in case of the given system was taken at 139℃ and temperature of condenser was kept unchanged at 20℃. For analysis the calculation of mass flow rate of the ystem was done through iterative method and system was analysed by maximising the first law efficiency with respect to the final pressure of the working fluid after compression.

Figure 4.3 represents the proposed system in which the simple ORC in the bottoming cycle will be replaced by the supercritical ORC system and the complete system will be analysed in a comparative manner to compare the output obtained from the Khaljani et.al. (2015) [32] with the proposed system.

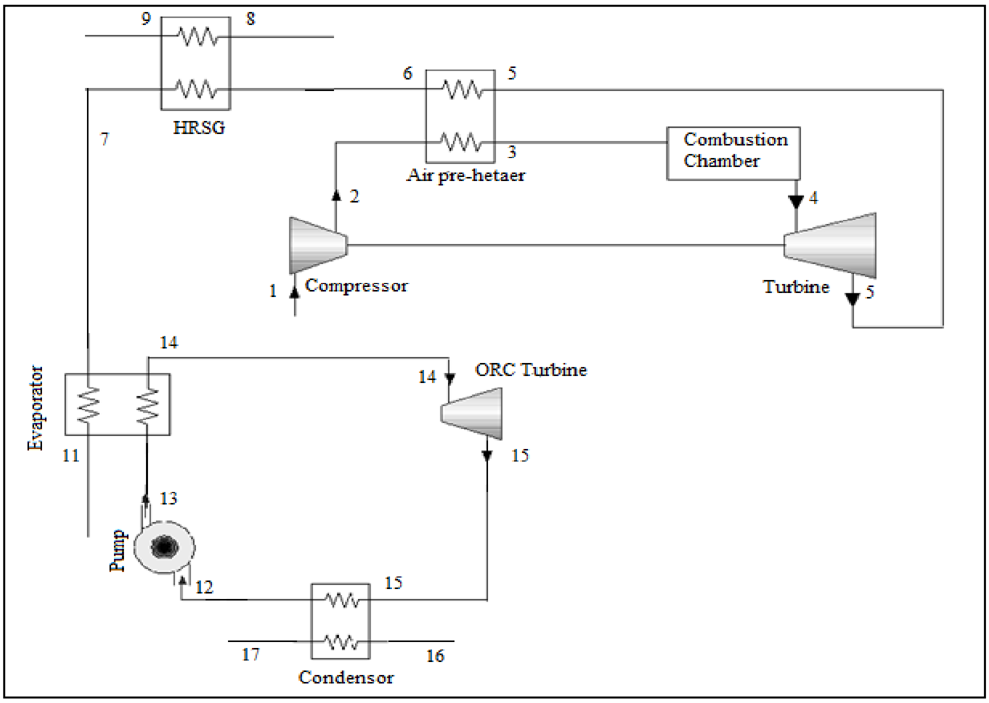


Figure 4.3 : Schematic diagram of GT-HRSG-supercritical ORC

4.3 THERMODYNAMIC MODELLING

Following assumptions were taken while modelling and analysing all the systems taken under consideration.

1. All the process are assumed to be a steady state process [32].
2. Adiabatic conditions are assumned for gas turbine and compressor [32].

3. For the fuel (methane) 50000Kj/Kg has been taken as lower calorific value. [32].
4. For combustion gasses and the air analysis concept of ideal gas was used [32].
5. 5% pressure drop on air side and 3% on the gasses side has been assumed [32].
6. Only 2% of the lower heating value has been transferred as the heat in combustion chamber. Rests of the systems are assumed to incur no heat loss [32].
7. The design of the HRSG system is such that t work on the sigle pressure mode only and the waters has a temperature of 298.15K and pressure of 35 bar when ir enters the system and it exits the HRSG in saturated state. [32]
8. The condition of the working fluid is saturated ar turbine inlet. [32].
9. In case of simple as well as supercritical ORC no pressure loss was considered in the system [30].

4.3.1 Thermodynamic Modelling of GT-HRSG-ORC system

Table 4.1 provides the equations of energy balance of the GT-HRSG-ORC cogeneration system and table 4.2 gives the exergy balance equations taken from Khaljani et.al.(2015) [32]. The equations are written considering the system shown in figure 4.1

Table 4,1: Energy balance equation of various component in GT-ORC cogeneration system
(Source : Reference [32])

Component	Energy Equation
Air compressor	$\eta_{ac} = \dfrac{(h_{2s} - h_1)}{h_2 - h_1} \quad \dot{W}_{AC} = m_{air}(h_2 - h_1)$
Air pre-heater	$h_3 - h_{2a} = (1 + \tau)(h_{5a} - h_6)$
Combustion chamber	$-0.02\tau LHV_{CH_4} + h_a + \tau h_p - (1 + \tau)h_p$
Gas turbine	$\eta_{GT} = (h_4 - h_5)/(h_4 - h_{5s}) \quad \dot{W}_{GT} = (\dot{m}_f + \dot{m}_{air})(h_4 - h_5)$
Pump	$\eta_p = \dfrac{v_{12}(p_{13} - p_{12})}{h_{13} - h_{12}} \quad \dot{W}_p = \dot{m}_{ORC}(h_{13} - h_{12})$
IHE	$(h_{14} - h_{13}) = (h_{16} - h_{17}), \in_{IHE} = (T_{11} - T_{14})/(T_{16} - T_{13})$
Evaporator	$(\dot{m}_{air} + \dot{m}_f)(h_7 - h_{11}) = \dot{m}_{ORC}(h_{15} - h_{14}), \dot{Q}_E = \dot{m}_{orc}(h_{15} - h_{14})$
Turbine	$\eta_T = (h_{15} - h_{16})/(h_{15s} - h_{15}) , \dot{W}_T = \dot{m}_{ORC}(h_{15} - h_{16})$
Condenser	$\dot{m}_{ORC}(h_{17} - h_{12}) = \dot{m}_{water}(h_{19} - h_{18}), \dot{Q}_{COND} = \dot{m}_{ORC}(h_{17} - h_{12})$

Table 4.2 : Exergy balance equations of various components in GT-ORC system (Source: Reference [32])

Component	Exergy Balance Equation
Air compressor	$\dot{E}_1 + \dot{W}_A = \dot{E}_2 + \dot{E}_{D,AE}$
Air pre-heater	$\dot{E}_2 + \dot{E}_5 = \dot{E}_3 + \dot{E}_6 + \dot{E}_{D,APH}$
Combustion chamber	$\dot{E}_3 + \dot{E}_{10} = \dot{E}_4 + \dot{E}_{D,CC}$
Gas turbine	$\dot{E}_4 = \dot{E}_5 + \dot{W}_{GT} + \dot{E}_{D,GT}$
Pump	$\dot{E}_{12} + \dot{W}_P = \dot{E}_{13} + \dot{E}_{D,P}$
IHE	$\dot{E}_{13} + \dot{E}_{16} = \dot{E}_{14} + \dot{E}_{17} + \dot{E}_{D,IHE}$
Evaporator	$\dot{E}_7 + \dot{E}_{14} = \dot{E}_{11} + \dot{E}_{15} + \dot{E}_{D,E}$
Turbine	$\dot{E}_{15} = \dot{E}_{16} + \dot{W}_T + \dot{E}_{D,T}$

4.3.2 Thermodynamic modelling of supercritical organic rankine cycle

The thermodynamic modelling of the supercritical ORC has been taken from Le et.al. (2014) [30] and equations are considering the diagram shown in the figure 4.2. Following is the energy analysis of various components of the system.

- Pumping process

$$\eta_{p,is} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{4.1}$$

$$\dot{W}_p = \dot{m}_{wf}(h_2 - h_1) \tag{4.2}$$

$$P_p = \dot{W}_p / \eta_{motor} \tag{4.3}$$

- High temperature heat transfer

$$\dot{Q}_{HTHEX} = \dot{m}_{wf}(h_3 - h_2) = \dot{m}_h(h_{hsi} - h_{hso}) \tag{4.4}$$

- Expansion Process

$$\eta_{t,is} = \frac{h_3 - h_{4,is}}{h_3 - h_4} \tag{4.5}$$

$$\dot{W}_t = \dot{m}_{wf}(h_3 - h_4) \tag{4.6}$$

$$P_t = \eta_{gen}\dot{W}_t \tag{4.7}$$

- Condensation Process

$$\dot{Q}_{cond} = \dot{m}_{wf}(h_4 - h_1) = \dot{m}_c(h_{cso} - h_{csi}) \tag{4.8}$$

Following is the exergy analysis of the components of the system :

Under steady state condition the exergy balance equation comes out as follows

$$\sum_j \dot{E}x_{q,j} - \dot{W}_{CV} + \sum_i \dot{E}x_i - \dot{I} = 0 \tag{4.9}$$

$$\sum_j (1 - \frac{T_o}{T_j}) . \dot{Q}_j - \dot{W}_{CV} + \sum_i \dot{m}_i Ex_i - \sum_e \dot{m}_e Ex_e - \dot{I} = 0 \tag{4.10}$$

Component wise the equation could be written as

- Pumping process

  Consumed Exergy

$$\dot{W}_p = \dot{m}_{wf}(h_2 - h_1) \tag{4.11}$$

  Useful exergy

$$\dot{E}x_p = \dot{m}_{wf}(h_2 - h_1 - T_0(s_2 - s_1)) \tag{4.12}$$

Exergy Destruction

$$\dot{I}_p = T_0 \dot{m}_{wf}(s_2 - s_1) \tag{4.13}$$

- High temperature heat transfer

Consumed Exergy

$$\dot{E}x_h = \dot{m}_h(h_{si} - h_{so} - T_0(s_{hsi} - s_{hso}) \tag{4.14}$$

Useful Exergy

$$\dot{E}x_{HTHEX} = \dot{m}_{wf}\left(h_3 - h_2 - T_0(s_3 - s_2)\right) \tag{4.15}$$

Exergy Destruction

$$\dot{I}_{HTHEX} = T_0\left[\dot{m}_{wf}(s_3 - s_2) + \dot{m}_h(s_{hso} - s_{hsi})\right] \tag{4.16}$$

- Expansion process

Consumed Exergy

$$\dot{E}x_t = \dot{m}_{wf}\left(h_3 - h_4 - T_0(s_3 - s_4)\right) \tag{4.17}$$

Useful Exergy

$$\dot{W}_t = \dot{m}_{wf}(h_3 - h_4) \tag{4.18}$$

Exergy Destruction

$$\dot{I}_t = T_0 \dot{m}_{wf}(s_4 - s_3) \tag{4.19}$$

- Condensation process

  Consumed Exergy

$$\dot{E}x_{cond} = \dot{m}_{wf}[h_4 - h_1 - T_0(s_4 - s_1)] \tag{4.20}$$

  Useful Exergy

$$\dot{E}x_c = \dot{m}_c\big(h_{cso} - h_{csi} - T_0(s_{cso} - s_{sci})\big) \tag{4.21}$$

  Exergy Destruction

$$\dot{I}_{cond} = T_0[\dot{m}_{wf}(s_1 - s_4) + \dot{m}_c(s_{cso} - s_{sci})] \tag{4.22}$$

Exergy balance equation for the supercritical ORC will give the complete exergy destruction through the system using following equation

$$\dot{E}x_h + \dot{W}_p = \dot{W}_t + \dot{E}x_c + \dot{I}_{tot} \tag{4.23}$$

Mass flow rate used in above equation is calculated for the system using an iterative method. For this process the pitch pint temperature difference for the evaporator is kept at 10 K. For this process the heat exchanger is first bifurcated into 100 parts for calculation of temperature at each section. In the iterative method certain value of mass flow rate is first assumed using this assmued mass flow rate, temperature at each section is determined and the difference in the temperature between the hot fluid and cold fluid is determined at each section and it is checked if the pitch point temperature difference condition is maintained or not. If the temperature goes bellow the pitch pint temperature difference then mass flow rate is decreased and if the difference in temperature obtained is higher than pitch point condition then the mass flow rate is increased and the calculation of temperature is done again. The process repeat itself till the condition for pitch point temperature difference is satisfied. Following equation represents the calculation used in iterative method for determining the mass flow rate of the system.

Temperature of working organic fluid at any section of heat exchanger is by

$$T_{wf,i+1} = T_{wf,i} + dT \tag{4.24}$$

where,

$$dT = (T_3 - T_2)/100 \tag{4.25}$$

So enthalpy of the hot fluid at the ith section and temperature difference at the section will be

$$h_{h,i+1} = h_{h,i} + \dot{m}_{wf}(h_{wf,i+1} - h_{wf,i})/\dot{m}_h \tag{4.26}$$

$$\Delta T_i = T_h - T_{wf,i} \tag{4.27}$$

The above obtained temperature difference is checked against the pitch point temperature difference required to be sustained in the evaporator and iteration continues till the condition is satisfied. Following diagram shows the flow chart for the iterative method used in determination of mass flow rate.

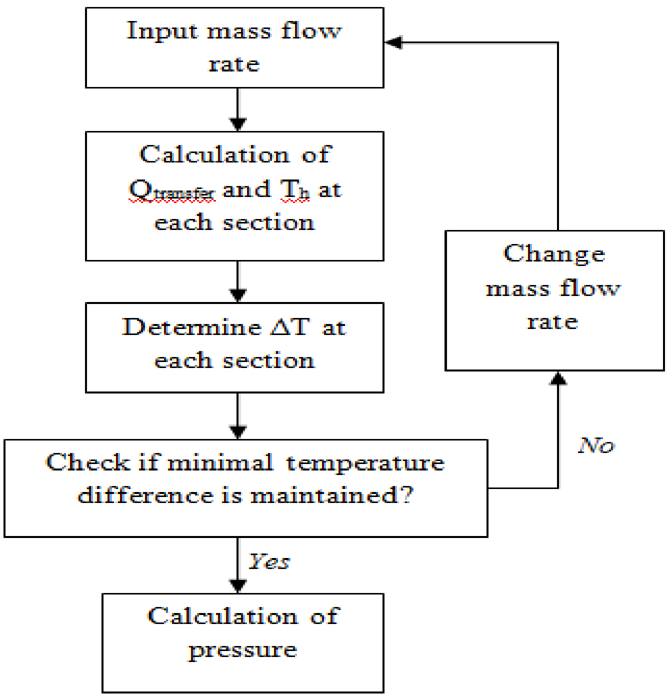


Figure 4.4: Flow chart for determination of mass flow rate in supercritical ORC (Source: Ref. [30])

Following graph represents the variation of temperature of air and the working fluid in supercritical ORC inside the evaporator. The diagram depicts the condition under which the pitch point temperature difference was maintained within the heat exchanger. For the purpose of validating the program R134a was used as the working fluid for generation of the following result as R134a was used in Le et.al.(2014) [30]. As it can be observed that near section 23 minimum temperature difference condition is attained for the system.

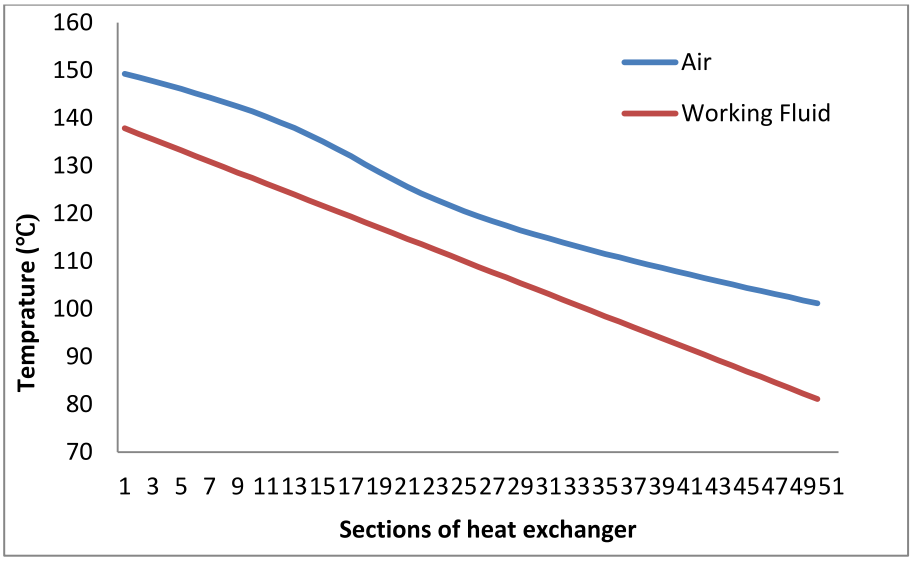


Figure 4.5: Temperature variation of Air and working fluid inside evaporator

4.4 MODEL VALIDATION

The two models of GT-HRSG-ORC and supercritical ORC were validated individually and then integrated into single system.

4.4.1 Model validation of GT-HRSG-ORC cogeneration system

For designing and validation of the model the basic parameters taken input for the system are given in table 4.3, these input parameters have same values as taken in khaljani et.al.(2015) [32].

Table 4.3: Parameters used in the modelling GT-HRSG-ORC cogeneration system (Source: Ref [32])

Parameters	Values
T1(K)	298.15
P1(bar)	1
$\dot{W}_{GT}$(MW)	30
T3(K)	850
T4(K)	1520

Table 4.3: Parameters used in the modelling GT-HRSG-ORC cogeneration system (continued)

r_p	10
$\eta_{AC}(\%)$	90
$\eta_{GT}(\%)$	80
$\Delta T_{pp,HRSG}(K)$	25
Te(K)	375
Tc(K)	303.20
$\Delta T_{pp,E}(K)$	10
$\eta_{T(ORC)}(\%)$	80%

The model was prepared in EES software and the properties of the working fluid obtained at every point were compared with the result obtained in the previous work khalijani et.al.(2015), the comparison is shown in table 4.4.

Table 4.4: Comparison of thermodynamic properties of each stream of GT- HRSG-ORC in present work and [32]

Sl. No	Fluid	T(K) [32]	T(K) This work	$\dot{m}$ (Kg/s) [32]	$\dot{m}$ (Kg/s) This work	$\dot{E}$(MW) [32]	$\dot{E}$(MW) This work
1	Air	298.15	298.15	94.75	95.06	0	0
2	Air	603.5	595.6	94.75	95.06	28.577	27.675
3	Air	850	850	94.75	95.06	43.540	42.486
4	Comb. gases	1520	1520	96.454	96.704	105.761	99.039
5	Comb. gases	1016	1018	96.454	96.704	41.947	38.166
6	Comb. gases	789.6	788.8	96.454	96.704	24.153	21.286
7	Comb. gases	422.1	422.2	96.454	96.704	3.604	2.414
8	Water	298.15	298.15	15.12	15.96	0.051	0.054
9	Water	515.7	515.7	15.12	15.96	14.833	15.660
10	Fuel	298.15	298.15	1.704	1.664	88.234	83.246
11	Comb. gases	381.5	380	96.454	96.704	2.448	0.9264
12	R123	303.2	303.2	21.61	21.41	0.0014	0.0014
13	R123	303.5	303.5	21.61	21.41	0.0120	0.0119
14	R123	316.2	316.2	21.61	21.41	0.0239	0.00237
15	R123	375	375	21.61	21.41	0.810	0.802
16	R123	323.8	323.8	21.61	21.41	0.079	0.0795
17	R123	305.5	305.6	21.61	21.41	0.063	0.0631
18	Water	298.15	298.15	177.1	173.7	0	0
19	Water	303.2	303.2	177,1	173.7	0.024	0.030

As it can be derived from above table that the program designed in EES works closely with the model designed in Khaljani et.al.(2015) [32]. Hence the model can be further explored for the parametric study.

4.4.2 Model validation of supercritical ORC system

Following table represents the input parameters which were taken in Le et.al.(2014) [30] for the analysis of supercritical ORC system.

Table 4.5: Parameters used in the modelling of supercritical ORC (Source: Ref. [30])

Parameters	Values
Isentropic efficiency for Pump and Turbine (%)	80
Efficiency of electrical generator and Pump motor (%)	90
Heat source / sink medium	Water
Heat source medium Inlet pressure (bar)	5
Heat source medium Inlet temperature (℃)	150
Heat source medium Mass flow rate (kg/s)	0.1
Heat sink medium Inlet Pressure (bar)	5
Heat sink medium Inlet Temperature (℃)	15

Table 4.6 and Table 4.7 compares the result obtained through EES programming with the result obtained in Le et.al. (2014) [30].

Table 4.6: Comparison of optimum pressure and mass flow rate obtained in model with [30]

Fluid	P (bar) [30]	P (bar) This work	m_{wf} (Kg/s) [30]	m_{wf} (Kg/s) This Work
R134a	55.86	56.02	0.1402	0.1392
R152a	50.08	50.34	0.0756	0.0742
R32	93.85	94.15	0.1234	0.124
R744	197.3	195.7	0.1848	0.1873
R290	60.49	60.46	0.0744	0.0745
R1234ze	47.2	47.32	0.1297	0.1284

Table 4.7: Comparison of first and second law efficiency and total exergy destruction in model with [30]

Fluid	η_I (%) [30]	η_I (%) This Work	η_{II} (%) [30]	η_{II} (%) This Work	$\dot{I}_{tot}$ (kW) [30]	$\dot{I}_{tot}$ (kW) This Work
R134a	14.0	14.17	57.7	59.14	3.4	3.64
R152a	14.9	15.05	57.2	58.62	2.9	3.025
R32	13.5	13.63	58.8	59.76	3.7	4.052
R744	10.7	10.87	49.7	50.08	4.7	5.07
R290	13.8	13.95	57.3	58.78	3.5	3.742
R1234ze	14.1	14.18	56.0	57.46	3.3	3.464

Results obtained in the table 4.6 and table 4.7 show that the model designed in EES software works closely with the model designed in the paper Le et.al. (2014) [30].

In this project the replacement of bottoming cycle of GT-HRSG-ORC cogeneration system was done by integrating supercritical ORC in with GT-HRSG system in place of simple ORC. The working fluid used in bottoming cycle for comparative and parametric study of the system is R134a. The critical point of R134a is established at 40.59 bar pressure and 374.21K temperature. Table 4.8 shows the properties at various points in the integrated system.

Table 4.8 Properties for GT-HRSG-supercritical ORC

Stream	Working Fluid	T(K)	P (bar)	ṁ (Kg/s)
1	Air	298.15	1.013	94.09
2	Air	595.6	10.13	94.09
3	Air	850	9.624	94.09
4	Comb. gases	1520	9.142	95.782
5	Comb. gases	1018	1.157	95.782
6	Comb. gases	788.8	1.122	95.782
7	Comb. gases	422.2	1.066	95.782
8	Water	298.15	35	13.41
9	Water	515.7	35	13.41
10	Fuel	298.15	12	1.692
11	Comb. gases	315.67	1.013	95.684
12	R134a	293.2	5.721	31.75
13	R134a	296.42	56.02	31.75
14	R134a	412	56.02	31.75
15	R134a	315.20	5.721	31.75
16	Water	288.15	5	167.8
17	Water	296.95	5	167.8

For parametric analysis of the system the change in the system behaviour was studies with respect to the following parameters

Table 4.9: Range of values taken for parametric study

Parameters	Values
Turbine Inlet Temperature (K)	1320-1520
Compression Ratio (bar)	5-10
Air pre heater Outlet Temperature (K)	850-1000
Air Inlet Temperature (K)	273-298
Gas turbine Efficiency (%)	85-95

The examination of exergy of the proposed system gives the following amount of exergy destruction in each of the component of the system.

Table 4.10: Exergy destruction of individual component for GT-HRSG-supercritical ORC system

Component	Exergy Destruction (KW)
Air Compressor	725.1
Air pre Heater	2051
Combustion Chamber	26692
Gas Turbine	1995
HRSG	3266
Evaporator	401.4
Supercritical ORC Turbine	295.5
Condenser	161
Supercritical ORC Pump	32.58

As it can be pointed out from above table, the maximum exergy destruction happens to be profoundly concentrated in the combustion chamber, reason being attributed to the highly irreversible combustion of fuel. It is followed by HRSG system which shows significant loss in exergy during the heat transfer process. The supercritical ORC equipment shows relatively less exergy destruction. After obtaining the properties and exergy destruction for individual component the system was subjected to parametric study according to the range of variation mention in table 4.9. The results obtained through the parametric analysis are discussed in the following chapter.

CHAPTER 5

RESULT AND DISCUSSIONS

Following are the results obtained through parametric analysis of the GT-HRSG-Supercritical ORC cogeneration system.

5.1 VARIATION WITH GAS TURBINE INLET TEMPERATURE (TIT)

5.1.1 Variation of first law efficiency with change in TIT

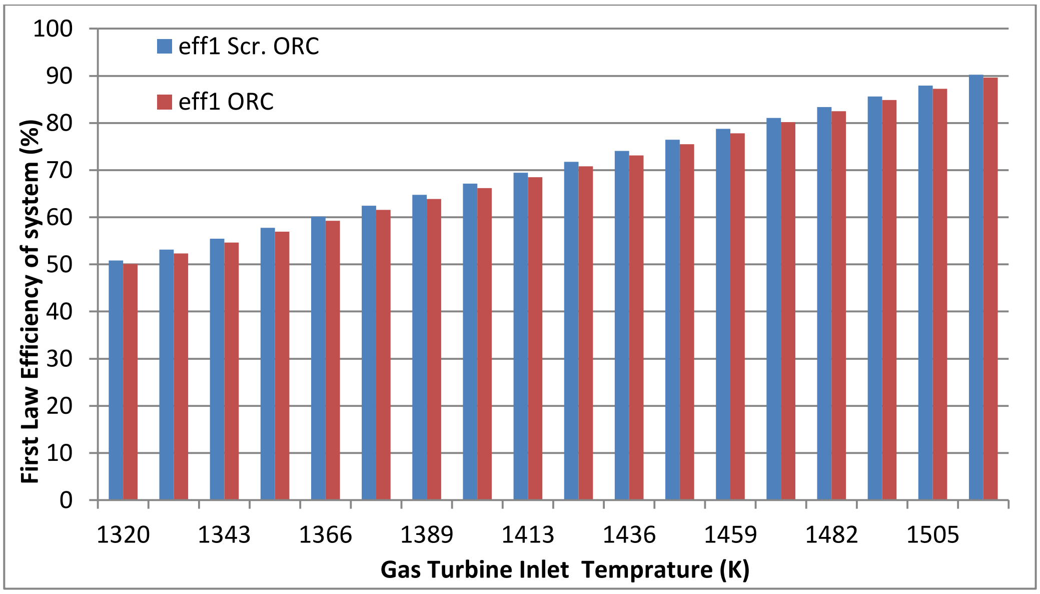


Figure 5.1: Variation of first law efficiency of the two system with change in TIT

As conclusion can be derived from above figure 5.1, that at all the temperature range taken the system with supercritical ORC performs better than system with simple ORC with IHE system. The system with supercritical ORC performs around 0.8 percentage point better than the system with simple ORC with IHE at lower temperature. It can also be observed that this difference in fist law efficiency in terms of percentage point first increases at reaches a maximum for the gas turbine inlet temperature around 1424K. After 1424K the difference in the first law efficiency of the two systems starts to decline sharply. As it can be perceived that the at higher values of turbine inlet temperature the performance of both the system reaches the same level. The figure 5.2 shows the variability of difference in the first law efficiency of the system with supercritical ORC from system with simple ORC with IHE in terms of percentage point.

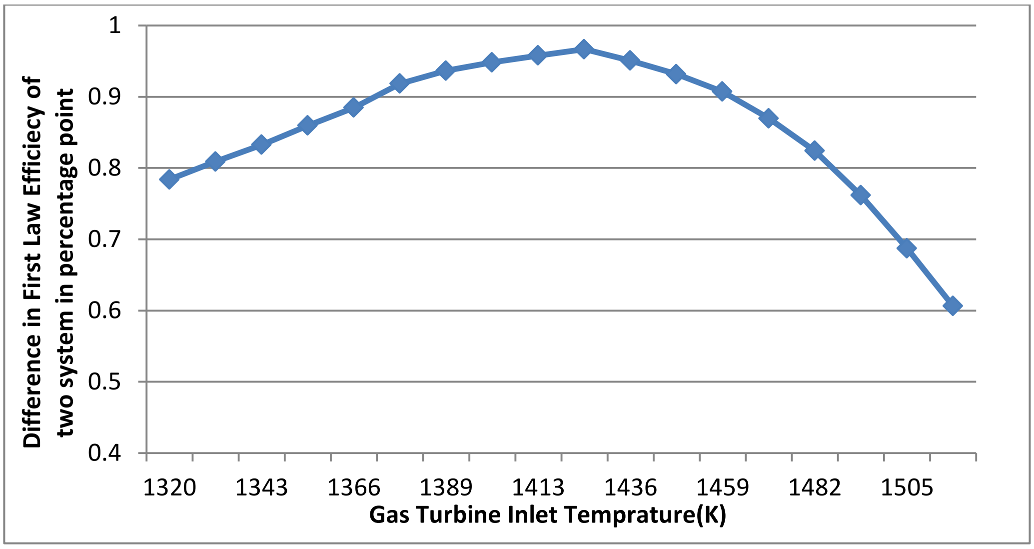


Figure 5.2: Variation of difference in first law efficiency of the two system with change in TIT

5.1.2 Variation of exergy efficiency with change in TIT

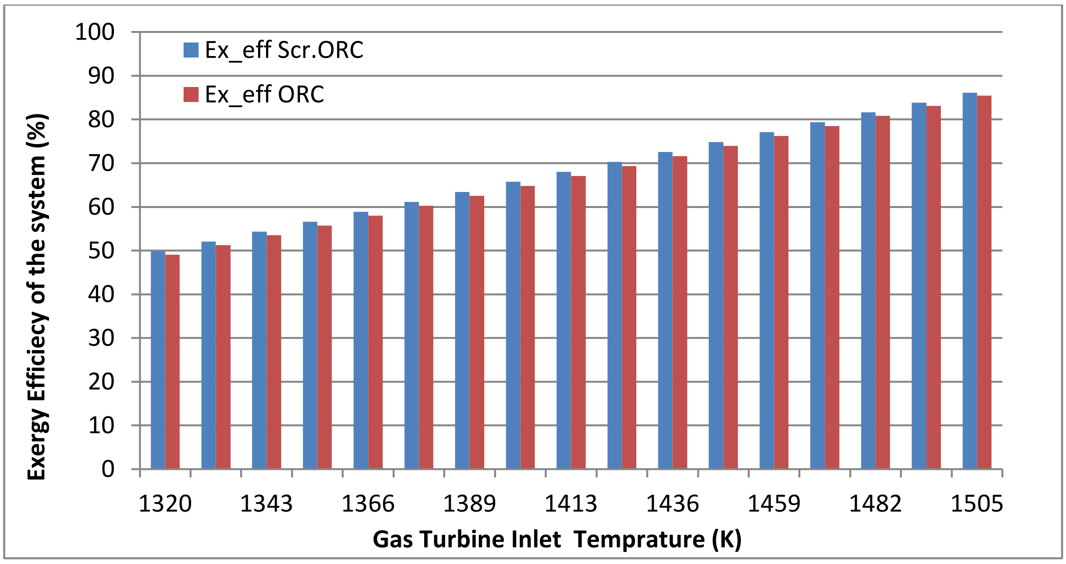


Figure 5.3: Variation of exergy efficiency of the two systems with TIT

From figure 5.3 it can be depicted that the system with supercritical ORC gives greater second law efficiency in compared to system with simple ORC with IHE. The pattern is similar to first law efficiency and the reason for the same can be attributed to the increased work output obtained through supercritical ORC which permits the operation of system at higher temperature and pressure condition without compromising the efficiency of the system. Again at lower range of turbine inlet temperature the system with supercritical ORC gives better performance when studied in terms of increased percentage point as depicted in figure 5.4. The difference in the exergy efficiency between the two systems attains the optimum point at 1413K after which it also shows decline. The maximum difference in the exergy efficiency of the two system is observed at, the system with supercritical ORC cycle performing 0.94 percentage point better than cycle with simple ORC and IHE.

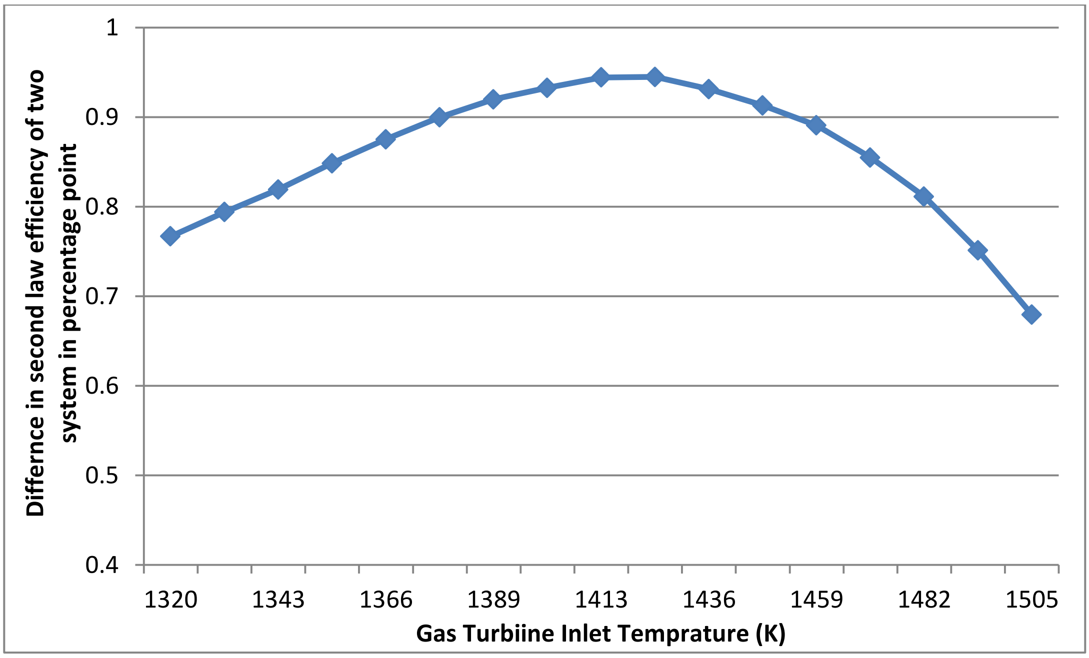


Figure 5.4: Variation of difference in exergy efficiency of the two system with change in TIT

5.1.3 Variation of exergy destruction ratio with change in TIT

For both the system the exergy destruction obtained is higher at lower range of TIT and goes on decreasing as the TIT goes on increasing. The relative study of the systems shows that the system with supercritical ORC has higher exergy destruction ratio in compared to system with simple ORC and IHE. This could be attributed to higher amount heat transfer associated with the organic fluid at high temperature and pressure condition in case of supercritical ORC which shows a greater irreversibility than the simple ORC with IHE. However as the TIT goes on increasing the difference in the exergy destruction of the two systems goes on decreasing. Figure 5.5 depicts the change in exergy destruction of the two systems with TIT.

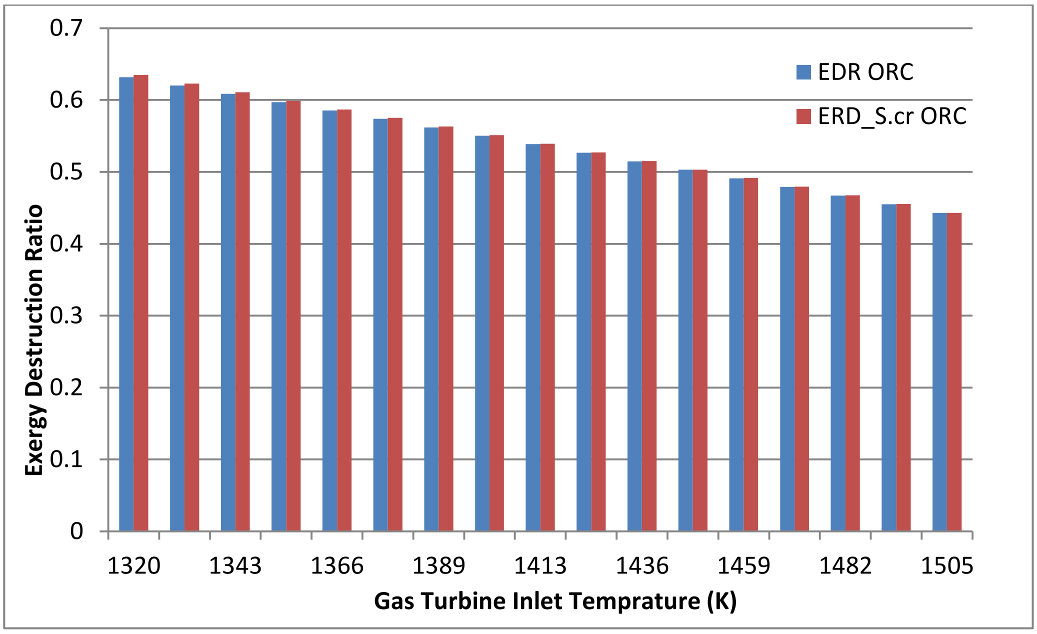


Figure 5.5: Variation in Exergy destruction ratio of the two systems with change TIT

The results obtained in the analysis of the system shows that the exergy destruction of the remaining system varies in same manner with temperature the only difference lies in the exergy destroyed in the bottoming ORC cycle. Figure 5.6 depicts the dissimilitude of exergy destroyed of two bottoming cycle with change in gas turbine inlet temperature.

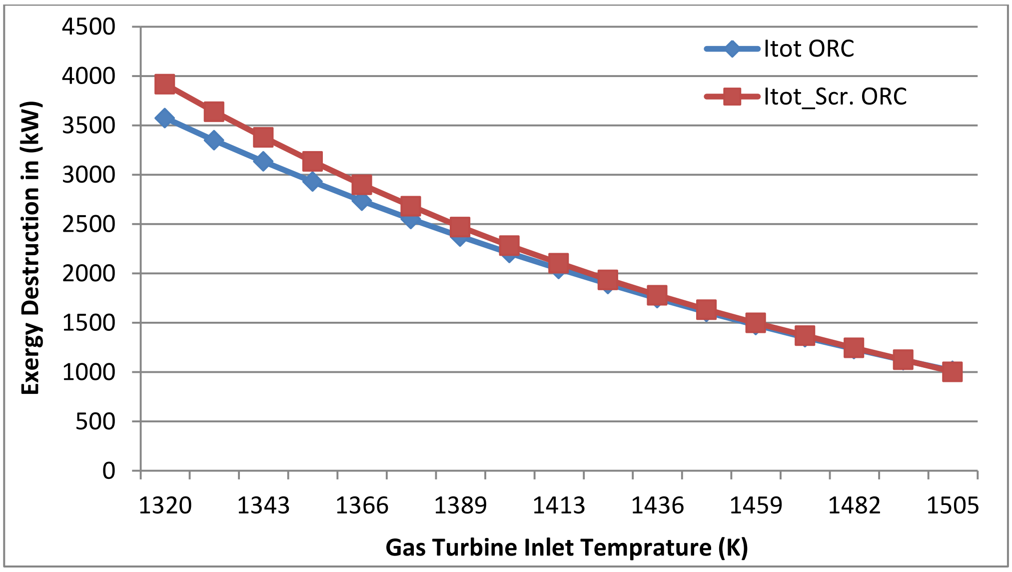


Figure 5.6 : Variation of exergy destruction of two bottoming ORC cycle with TIT

The variation in the figure 5.6 clearly provides that the exergy destruction of supercritical ORC system was higher than the exergy destruction of the simple ORC system with IHE and as the TIT goes on increasing the difference between the exergy destruction of the two systems goes on decreasing and become almost same for higher values of TIT. This explains the behaviour of EDR with respect to TIT.

5.1.4 Variation of bottoming cycle work output with change in TIT

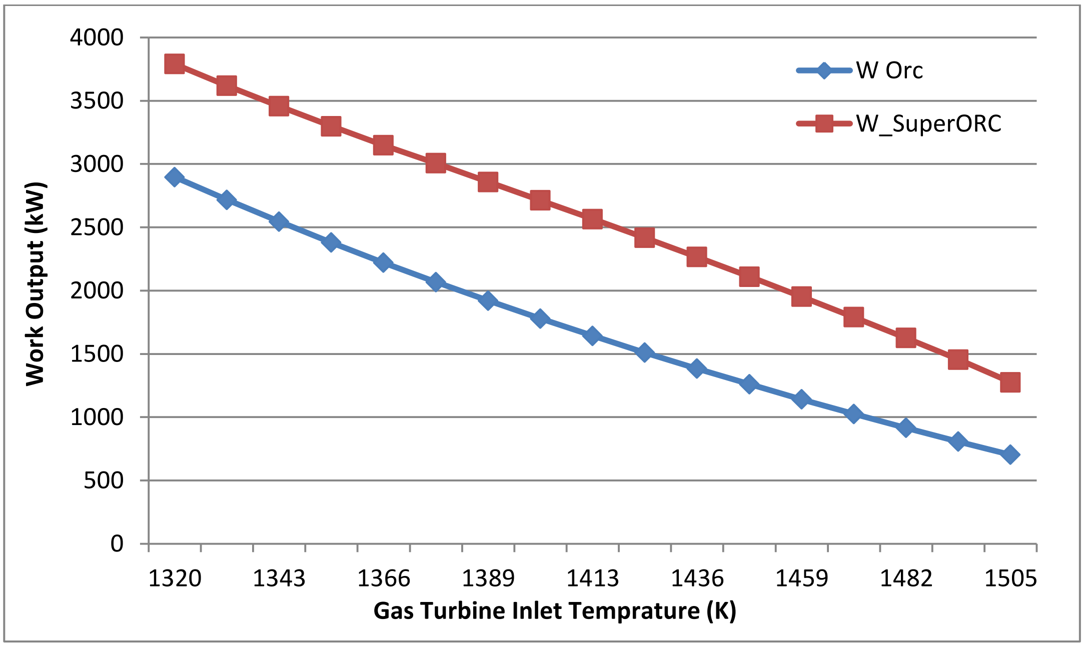


Figure 5.7: Variation of work output of bottoming ORC with change in TIT

As it can be deduced that with increment in the TIT the work output of both the system goes on decreasing. The decrease can be attributed to decease in the outlet temperature of the HRSG as a result of increase in the amount of amount of process heat generated through HRSG and decrement in the flow rate of air. However, at lower values of TIT higher ORC output helps in maintaining the overall efficiency of the system when the comparatively the heat recovered from the HRSG system is lower. This justifies the use of ORC in the bottoming cycle especially for operating turbine and comparatively lower temperature. In comparative study it can be concluded that the work output of the supercritical ORC remain significantly higher than the work output of simple ORC with IHE. The decrease in the work output of supercritical ORC is comparatively linear while rate of decrease is higher for lower values of TIT in case of simple ORC cycle with IHE and this rate of decrease goes on decreasing with increase in TIT. The higher amount of waste heat recovery for all values of TIT can justify the use of supercritical ORC in place of simple ORC in bottoming cycle.

5.2 VARIATION WITH GAS TURBINE COMPRESSION RATIO (r)

5.2.1 Variation of first law efficiency of systems with 'r'

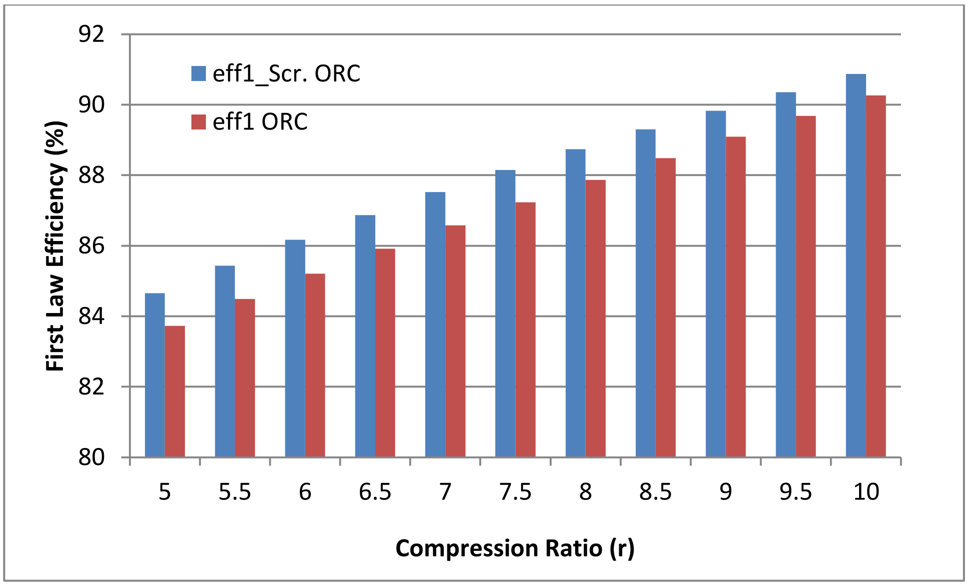


Figure 5.8 : Variation of first law efficiency with change in compression ratio

As shown in the figure 5.8, as the compression ratio of the system increases the first law efficiency of both the system goes on increasing. However, it can be observed that the system with supercritical ORC performs better than the system with simple ORC with IHE as bottoming cycle for all the range if compression ratio.

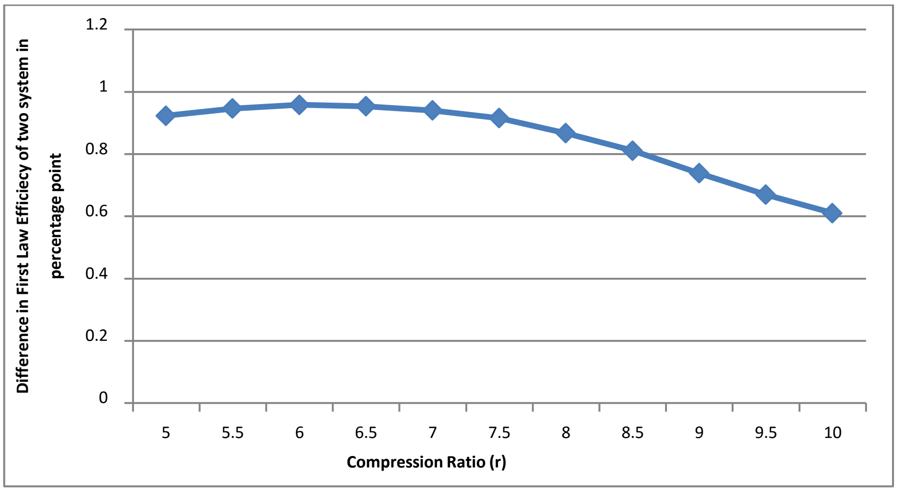


Figure 5.9: Variation of difference in first law efficiency of the two system with change in 'r'

The effect of better performance is more profound at lower compression ratio and as the compression goes on increasing difference between the performances of the two systems goes on decreasing. Figure 5.9 shows the difference in terms of percentage point between the first law efficiency of the two systems. It can be noted that at lower compression ratio of gas turbine the system with supercritical ORC performs as much as 0.95 percentage point better than the system having simple ORC with IHE. From the results obtained it can also be concluded that for getting a desired performance from the system the system with supercritical ORC can be operated at comparatively lower compression ratio of gas turbine that will be helpful in fulfilling the design requirement of the system and can also reduce the cost of manufacturing the component.

5.2.2: Variation of exergy efficiency of systems with 'r'

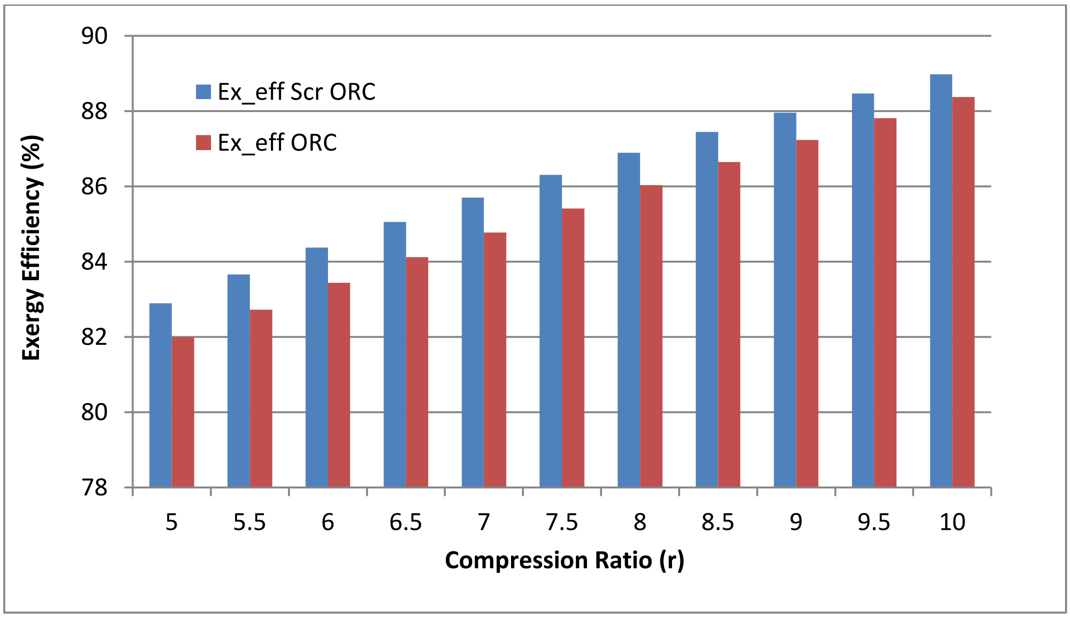


Figure 5.10: Variation of exergy efficiency of the systems with change in 'r'

The exergy efficiency also shoes the similar variation as the first law efficiency. For both the system the exergy efficiency goes on increasing as the compression ratio goes on increasing and it can be noted from the figure 5.10 that system having supercritical ORC performs better than the system having simple ORC with IHE as the bottoming cycle for entire range of compression ratio. The comparative study of the two system shows that system with supercritical ORC as bottoming cycle can enhance the overall efficiency of the system from 0.6 percentage oint to 0.93 percentage point. At lower compression ratio the amount of percentage point by which the system with supercritical ORC outperforms the system having simple ORC with IHE as the bottoming cycle is higher is higher at lower compression ratio. So, as the compression ratio goes on increasing the corresponding difference in the performance of the two system goes on decreasing. As shown in the figure this difference attain an optimum point at a compression ratio of 6.5 at which the system with supercritical ORC performs 0.938 percentage point better than the system having simple ORC with IHE as bottoming cycle.

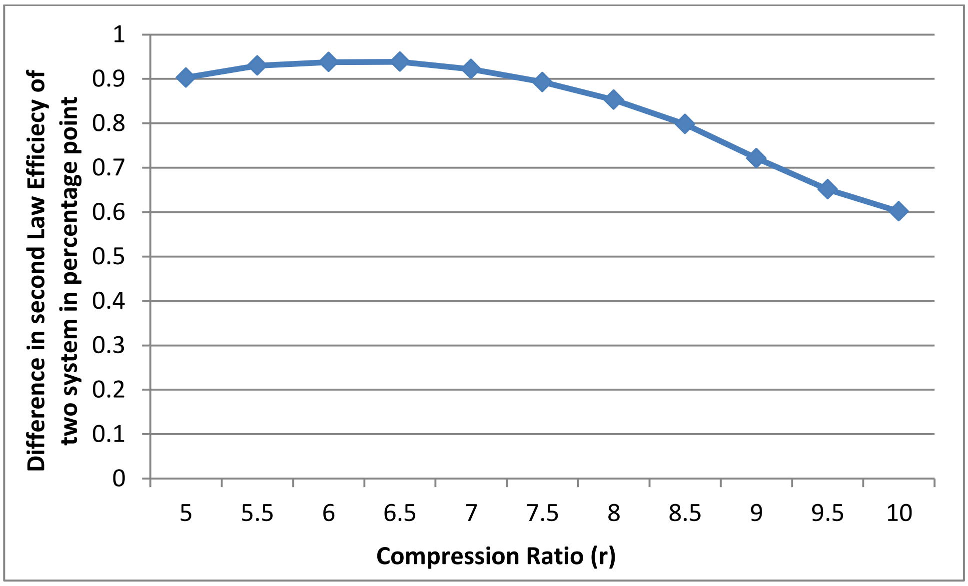


Figure 5.11: Variation of difference in exergy efficiency of the two system with change in ‘r’

5.2.3 Variation of exergy destruction ratio with change in ‘r’

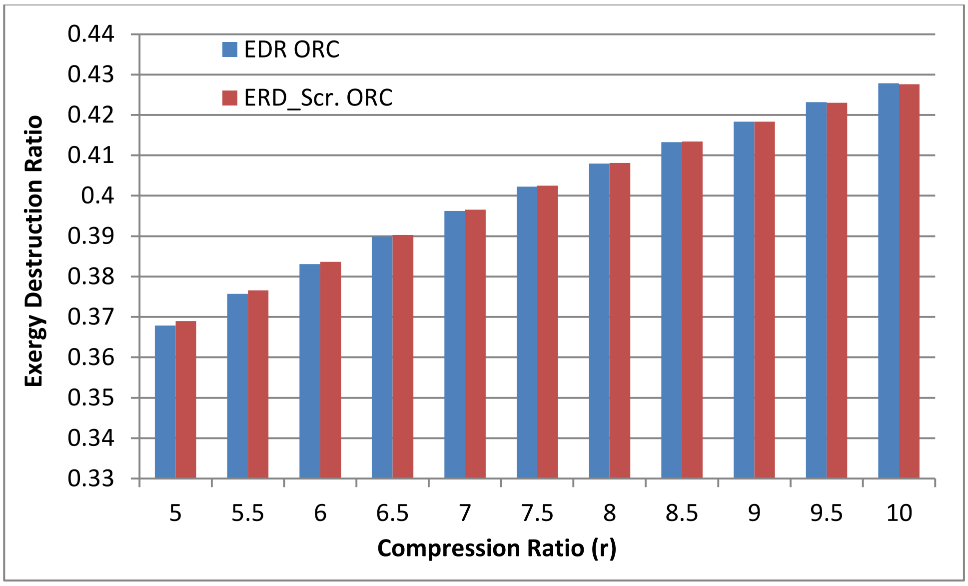


Figure 5.12: Variation of exergy destruction with change in ‘r’

Figure 5.12 shows that with increment in the compression ratio the exergy destruction of both the system goes on increasing. The comparative study between the two system concludes that at lower compression ratio the system with supercritical ORC as bottoming cycle shows slightly higher exergy destruction in compared to the system having simple ORC with IHE as the bottoming cycle. However, at higher compression ratio the two system has almost same exergy destruction ratio. The analysis depicts that the exergy destruction of the rest of the system remains unaffected by the change in bottoming cycle. So for comparative analysis, the exergy loss in overall system can be understood by analysis the exergy lost in the corresponding bottoming cycle. Figure 5.13 depicts that the exergy destruction of the two bottoming cycle. It can be seen that the variation completely resembles the change in complete exergy destruction of the system. The exergy destruction of the bottoming cycle shows that supercritical ORC has higher exergy losses at lower compression ratio than ORC with IHE and with increase in the compression ratio the exergy destruction of both the cycle goes on decreasing however, the rate of decrease in case of supercritical ORC cycle is higher, as it can be seen from graph that the line corresponding to it has higher negative slope. As a result the exergy destruction of the two systems becomes equal near to compression ratio of 7.5 to 8.5.

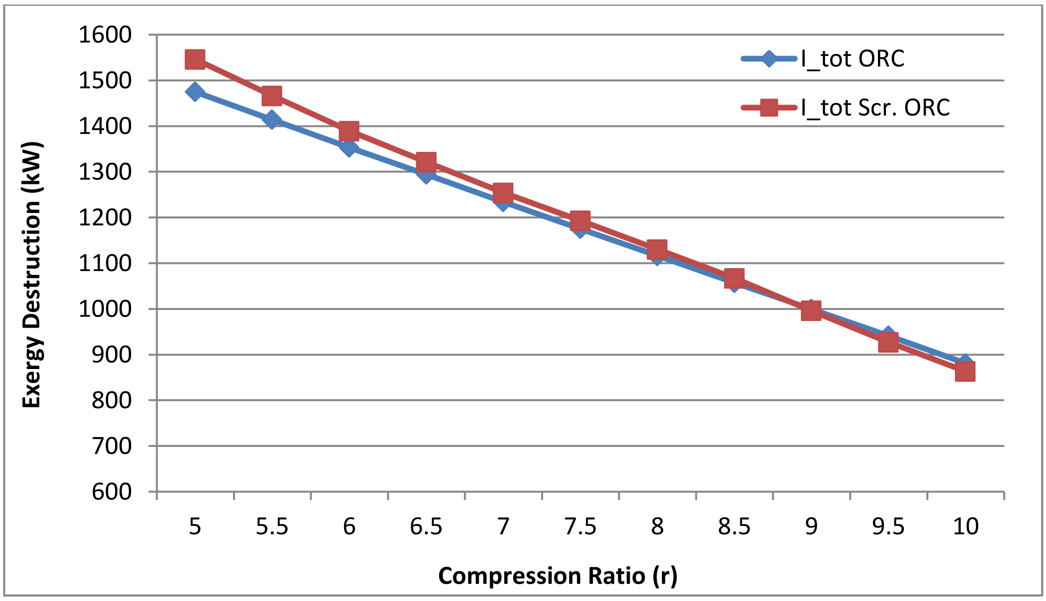


Figure 5.13: Variation of exergy destruction of two bottoming ORC with 'r'

Based on the comparative examination of the exergy destruction the utilisation of supercritical cycle for compression ratio of above 7.5 is recommended. However, for compression ratio above 7 the performance of supercritical ORC goes on deteriorating in compared to the simple ORC as shown in case of first law efficiency and exergy efficiency.

5.2.4 Variation of Work done in bottoming cycle with change in 'r'

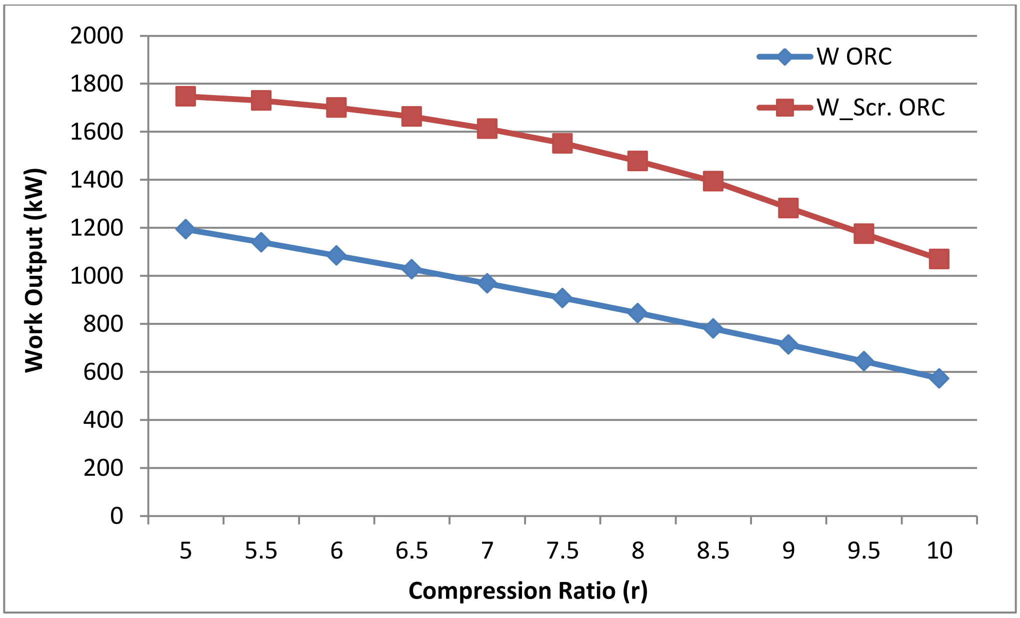


Figure 5.14: Variation of work output of two bottoming ORC with 'r'

From Figure 5.14 it can be noted that with increase in the compression ratio the work throughput of both the system goes on decreasing. This relation is in contrary to the increased in efficiency obtained in case of first law analysis and exergy analysis of the system. This could be attributed to increase in heat extracted in the from HRSG system. As compression ratio increases the heat extracted from HRSG system also goes on increasing, as shown in figure 5.15, as a result of which the amount of waste heat available for recovery from the ORC system goes on decreasing and therefore the work of the two system goes on decreasing.

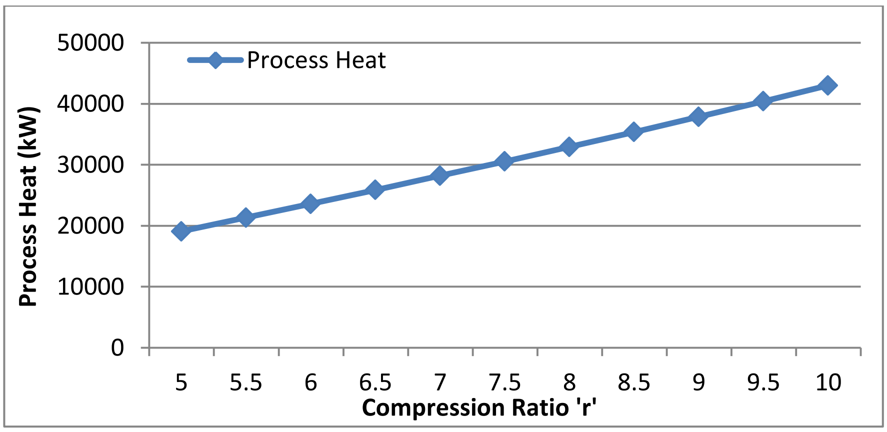


Figure 5.15: Variation of process heat with change in 'r'

The comparative study shows that the work output of the system with supercritical ORC as bottoming cycle remains comparatively higher than system having simple ORC with IHE as bottoming cycle. This result reflects on both first law analysis and second law analysis of the system. It can also be noted from the figure that the rate of decrease in case of simple ORC with IHE is linear in compared to supercritical ORC cycle, in which the rate of decrease is lower for initial values of compression ratio and increases with increase in the compression ratio.

5.3 VARIATION WITH APH OUTLET TEMPERATURE

5.3.1 Variation of first law efficiency with APH outlet temperature

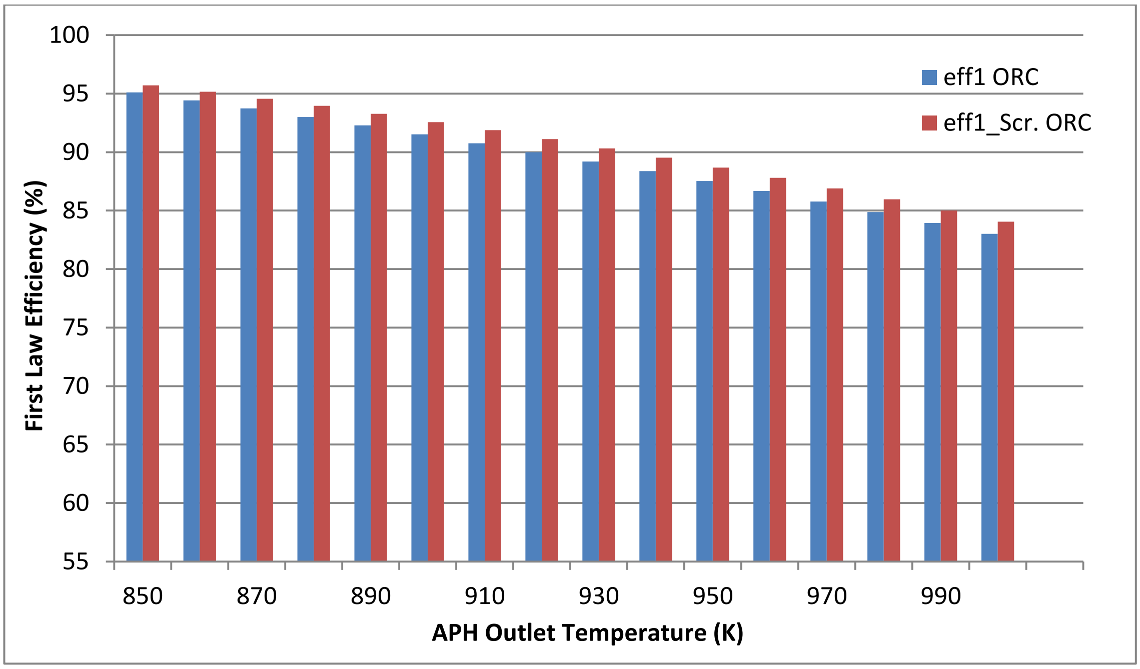


Figure 5.16: Variation of first law efficiency with APH outlet temperature.

The first law efficiency of the system goes on decreasing as the APH outlet temperature increases for both the system. Although the mass flow rate of the fuel shoes decrement with increment in APH outlet temperature but the amount of heat extracted in the form of process heat through HRSG system also goes on decreasing. The reason for this decrease can be attributed to decreased temperature at APH on the exhaust gas side. This decrease in the process heat has a dominating influence on the system and cause the decrease in the overall efficiency of the system. The comparative study between the system depicts that the rate of decrease of the performance is higher in case of system having simple ORC with IHE as the bottoming cycle. As the difference between the first law efficiency of the systems with supercritical ORC and system having simple ORC with IHE as bottoming cycle, in terms of the percentage point, initially goes on increasing with increment in APH outlet temperature and decreases slightly at higher level of APH outlet temperature as shown in figure 5.17.

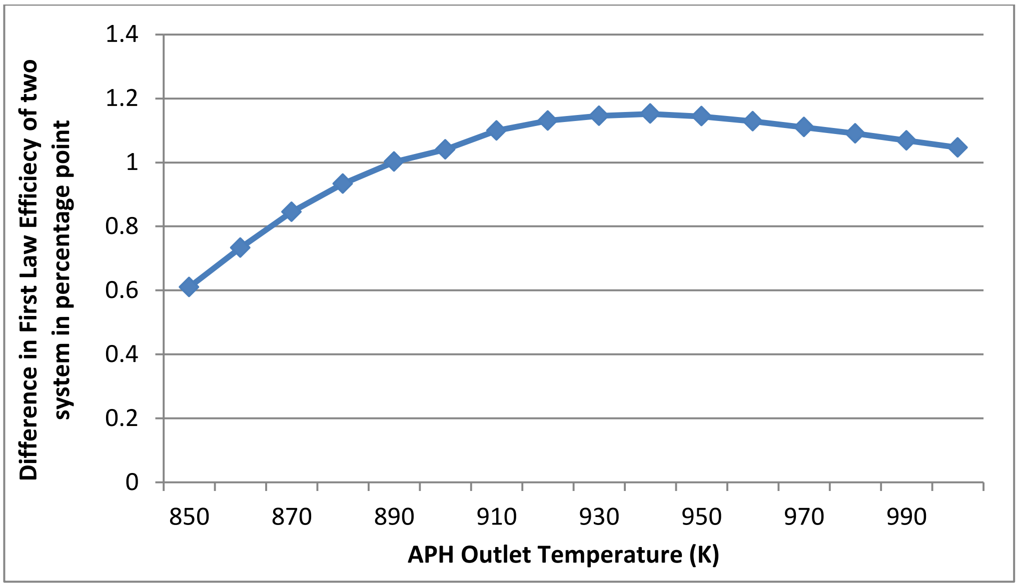


Figure 5.17: Variation of difference in first law efficiency of the two system with change in APH outlet Temperature

As it can be noted that for initial level of APH inlet temperature the difference between the efficiencies of the two systems is low but as it increases the difference also goes on increasing. At the temperature value of 940 K this difference reaches its maximum point where the system with supercritical ORC performs 1.2 percentage point better than the system having ORC with IHE as bottoming cycle. As the temperature further increases the difference of the efficiency remain rather stable and no rapid decrease in observed. However the first law efficiency of the system get lowered by around 5 percentage point near temperature 940 K which creates a contradicting and don't justify the use of supercritical ORC in proper manner.

5.3.2 Variation of Exergy efficiency with change in APH outlet temperature

The exergy efficiency shows the same variation as the first law efficiency i.e., the exergy efficiency of both the system goes on decreasing with increment in the APH outlet temperature. It can be noted that the exergy efficiency is lower in case of system having ORC with IHE in compared to the system having supercritical ORC as bottoming cycle. Figure 5.18 shows the comparative study of the two systems.

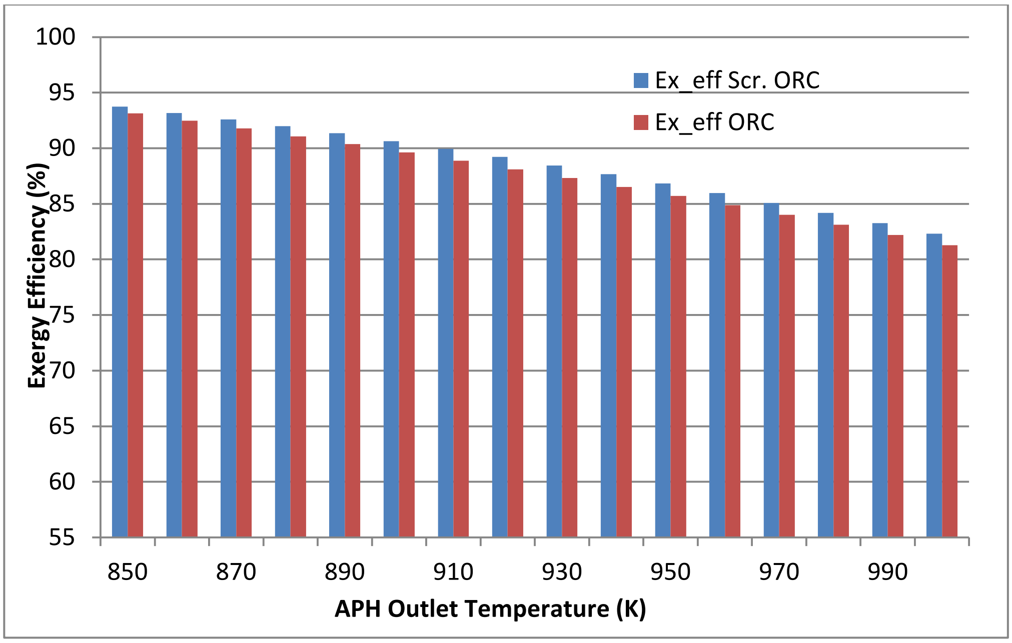


Figure 5.18: Variation of exergy efficiency with APH outlet temperature.

Figure 5.19 provide the insight of the difference between the exergy efficiency of the two systems.

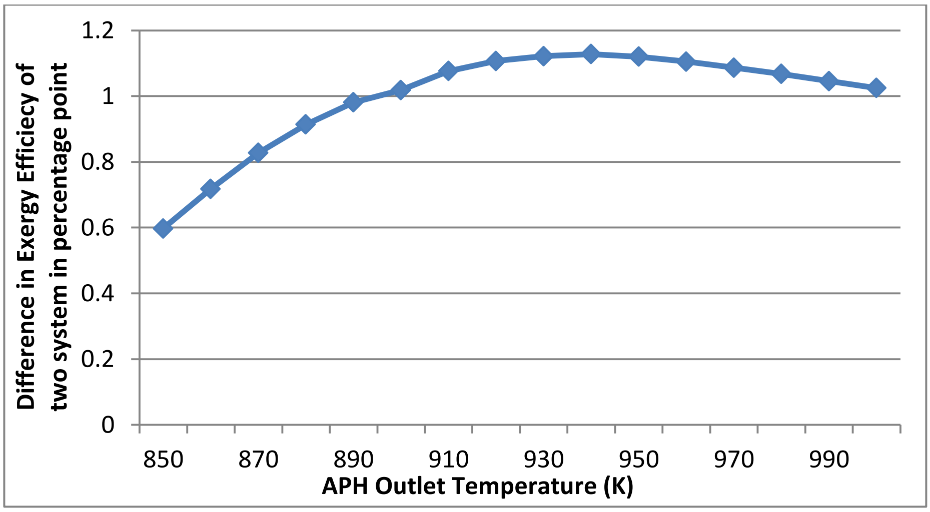


Figure 5.19: Variation of difference of the exergy efficiency of two system with change in APH outlet temperature

As it can be depicted from the figure 5.19 at initial level of APH outlet temperature the supercritical cycle performs only 0.6 percentage point better than system having ORC with IHE. As the APH outlet temperature goes on increasing the difference between the two system rises rapidly and attain the maximum neat 940 K at which the system with supercritical ORC performs around 1.1 percentage point better than the system having simple ORC with IHE as bottoming cycle. The variation of the difference in case of first law efficiency and in case of second law efficiency is same as for both the cases the change in numerator remains the same where as the decrease in the exergy supply and energy supply shows almost the same variation.

5.3.3 Variation of exergy destruction ratio with change in APH outlet temperature

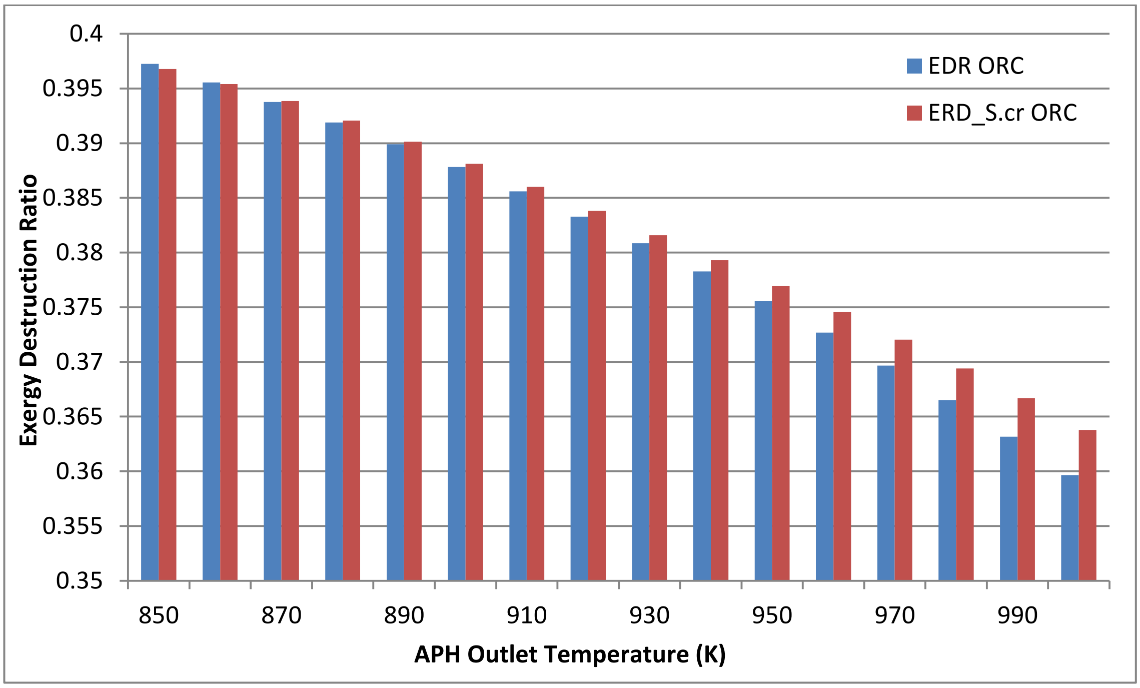


Figure 5.20: Variation of exergy destruction ratio with AHP outlet temperature

Figure 5.20 provides the change of exergy destruction ratio with change in APH outlet temperature. It can be observed that with increment in the outlet temperature of APH the exergy destruction ratio for both the system goes on decreasing. The reason for the same can be attributed to decrease in the exergy loss from combustion chamber, which cause the major amount of exergy destruction, as a result of decreased mass flow rate of fuel. The comparative study shows that for lower APH outlet temperature the exergy destruction obtained from system with supercritical ORC is less in compared to the system having simple ORC with IHE. For the range of 870 K to 890 K the exergy destruction remains almost equal for both the system. However as the APH outlet temperature further increases, the system having simple ORC with IHE shows higher rate of decease in compared to system with supercritical ORC. This variation in rate of decrease can be explained through exergy destruction of individual bottoming cycle. Figure 5.21 depicts the change of exergy destruction of supercritical ORC and simple ORC with IHE in the bottoming cycle with change in APH outlet temperature.

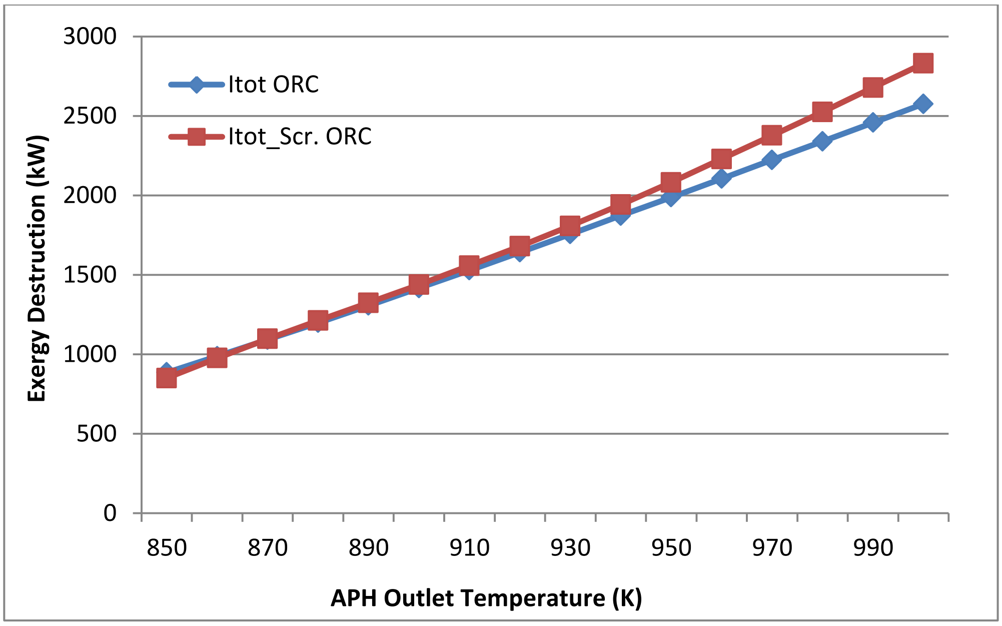


Figure 5.21: Variation of exergy destruction of bottoming cycle with change in APH outlet temperature

From the figure 5.21 it is very much clear that as the APH outlet temperature increases the exergy destruction of the two bottoming cycle goes on increasing. This increment in the exergy destruction of the two system can be attributed to increase in the heat transfer that occurs in the evaporator due to decrement in the performance of the HRSG system. The comparative study between the two system shows that at lower levels of APH outlet temperature the exergy destruction of the two bottoming cycle is equal however with rise in APH outlet temperature the exergy destruction of the supercritical ORC increases at a faster rate in compared to the simple ORC with IHE. This explains the relatively higher rate of decrement in total exergy destruction ratio of the system having simple ORC with IHE in compared to the system with supercritical ORC as bottoming cycle as the rapid increment in the exergy destruction of supercritical ORC counteracts to the decrement in the exergy destruction achieved in rest of the system. Although with increase in APH outlet temperature there is decrease in the exery destruction ratio which is a favourable outcome of the system still increased APH outlet temperature effect the first law efficiency and exergy efficiency in a negative way. Hence the system demands an optimisation, according to the desired characteristics required from the system.

5.3.4 Change in the work output of bottoming cycle with change in APH outlet temperature

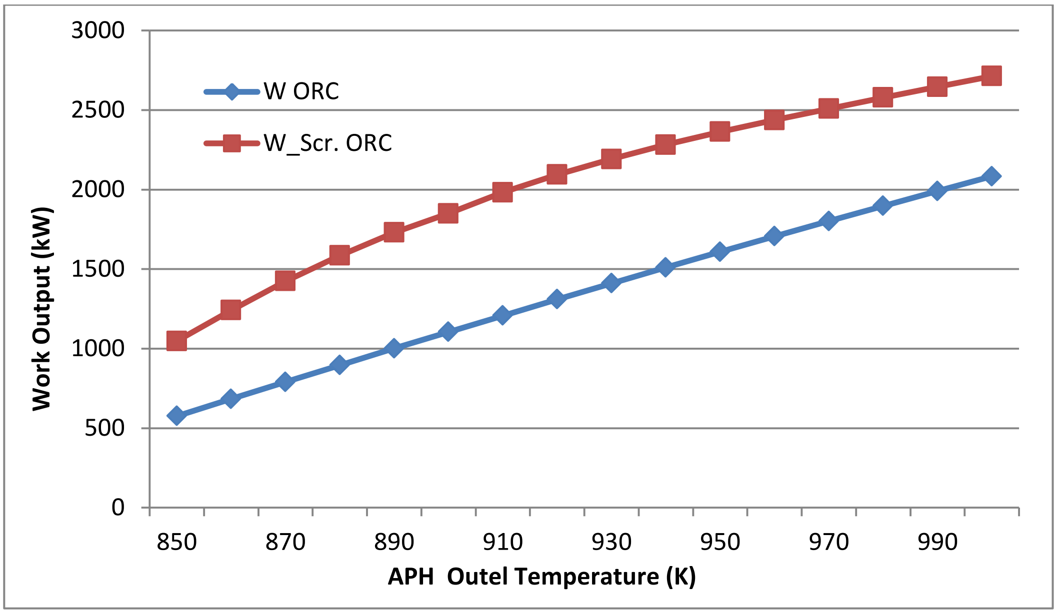


Figure 5.22: Variation of work output of bottoming cycle with APH outlet temperature

As depicted in figure 5.22, with increment in the APH outlet temperature the work output of the two systems goes on increasing. This increment in the work output of the bottoming cycle can be attributed to decreased performance of HRSG system i.e., the amount of process heart extracted from the HRSG system goes on decreasing as the APH outlet temperature goes on increasing. This leads to more amount of heat available for recovery from the bottoming cycle as a result the work output of the bottoming cycle goes on increasing in both the cases. The comparative study of the two system shows that the work execrated form the waste is higher in case of supercritical ORC in compared to simple ORC with IHE. The variation also shows that for initial values of APH outlet temperature the rate of increase of work output in case of supercritical ORC is higher but this rate goes on decreasing with increase in APH outlet temperature. In contrast to that the rate of increase of work output in case of simple ORC with IHE remains constant. As a result the difference between the work output of the two system first goes on increasing with increase in APH outlet temperature and the decreases after attaning a maximum point. This variation in the difference in the work output is reflected very well in the change in first law efficiency and exergy efficiency with change in APH outlet temperature.

5.4 VARIATION WITH AIR INLET TEMPERATURE

5.4.1 Variation of first law efficiency with change in air inlet temperature

Figure 5.23 depicts the variation of system first law efficiency with change in the air inlet temperature. It can be noted from the graph that with increment in the air inlet temperature the net work output of both the systems goes on increasing.

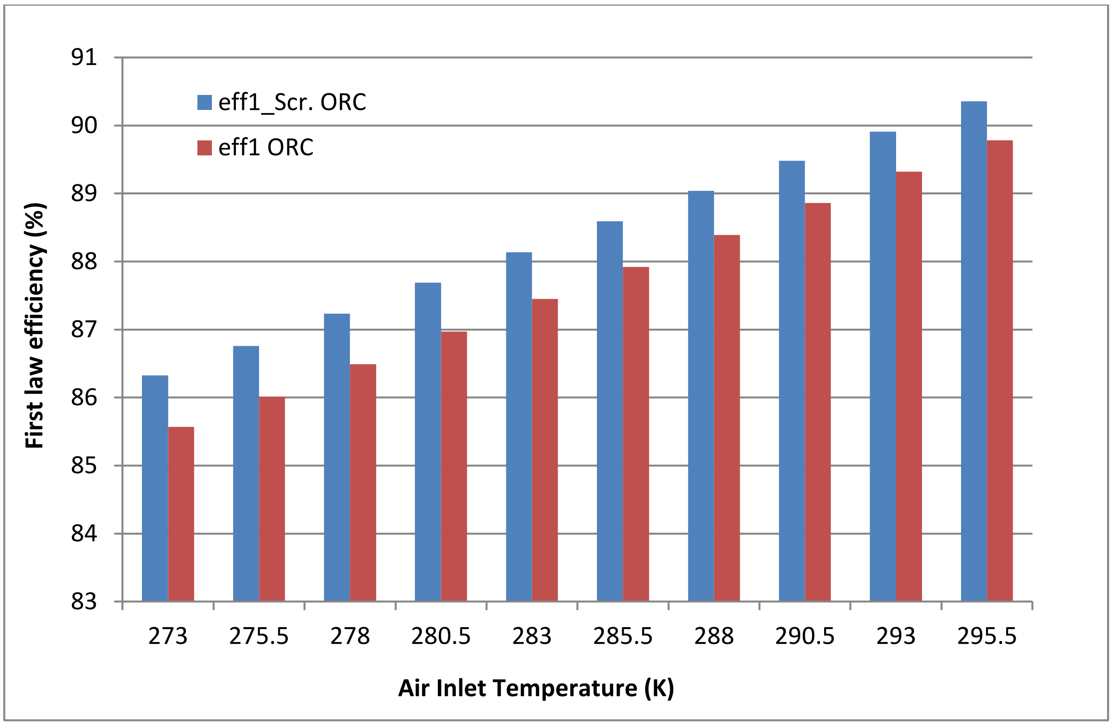


Figure 5.23: Variation of first law efficiency with change in air inlet temperature

It is interesting to observe that the increment in the inlet air temperature couses the work consumed by the compressor to increase, this leads to enhanced mass flow rate to meet the requirement of 30MW of power production through gas turbine system. As a result of enhanced mass flow rate, the heat recovered though HRSG system goes on increasing. This increased output in the form of process heat compensate for both increased work done on compressor as well as decreased output at bottoming ORC cycle. Hence the first law efficiency in both the cases goes on increasing. Now, the comparative study of heat recovered through two bottoming cycle shows that, the system having supercritical ORC has better performance than the system having ORC with IHE throughout the range of atmospheric temperature. It can also be observed that the difference in the first law efficiency at lower atmospheric pressure is higher in compared to the difference in the first law efficiency at higher atmospheric temperature. The reason can be attributed to differential rate of increase, which is observed to be higher in case of system having simple ORC with IHE. Figure 5.24 depict the variation of difference between the first law efficiency of the two systems with change in air inlet temperature.

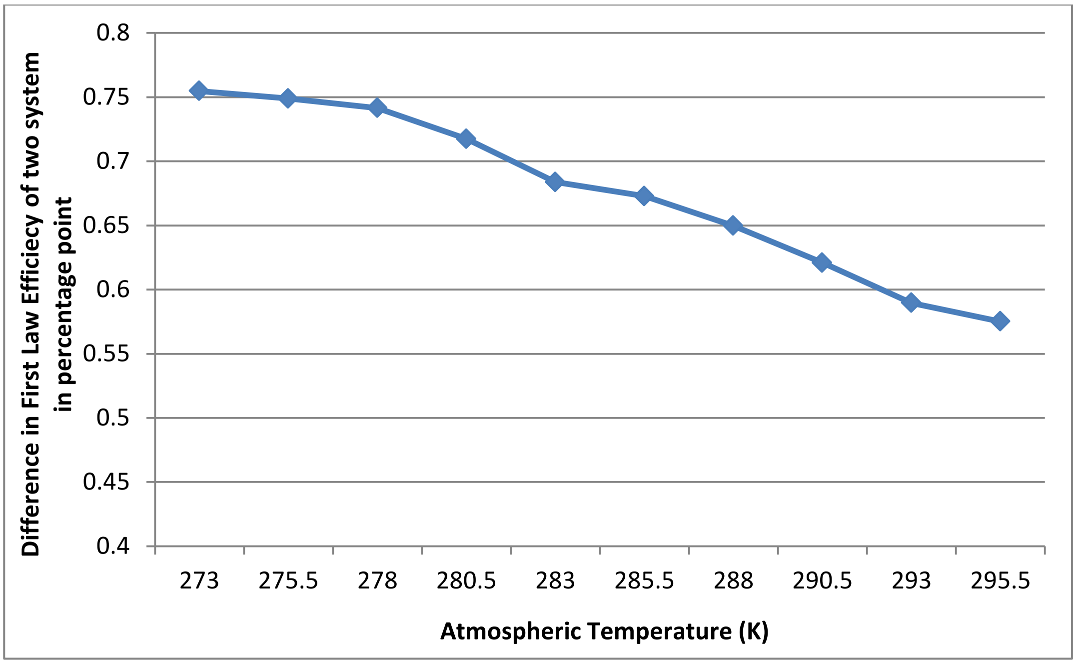


Figure 5.24: Variation of difference in first law efficiency with air inlet temperature

Better performance of the system having supercritical ORC in bottoming cycle is obtained at lower air inlet temperature. As it can be observed that it performs 0.75 percentage point better than system having ORC with IHE as bottoming cycle. After which this difference continuously goes on decreasing. It can be deduced from above analysis that if the cogeneration system is going to be operated at a comparatively lower temperature it would be favourable for the system to be operated with supercritical ORC as bottoming cycle as it will favour higher heat energy extraction form the exhaust gasses. .

5.4.2 Variation of exergy efficiency with change in air inlet temperature

The variation of the exergy efficiency is very similar to the variation in the first law efficiency. For bath the system the exergy efficiency goes on increasing with increment in the atmospheric temperature. The cause for similar variation is attributed to the fact that exergy supplied variation in case of exregy efficiency varies similar to energy supplied due to its dependency directly on the mass flow rate of air only, which is the only parameter of the exergy supply which is varying in this analysis. The comparative study of the two systems can also be carried out based on the analysis of figure 5.25 and figure 5.26.

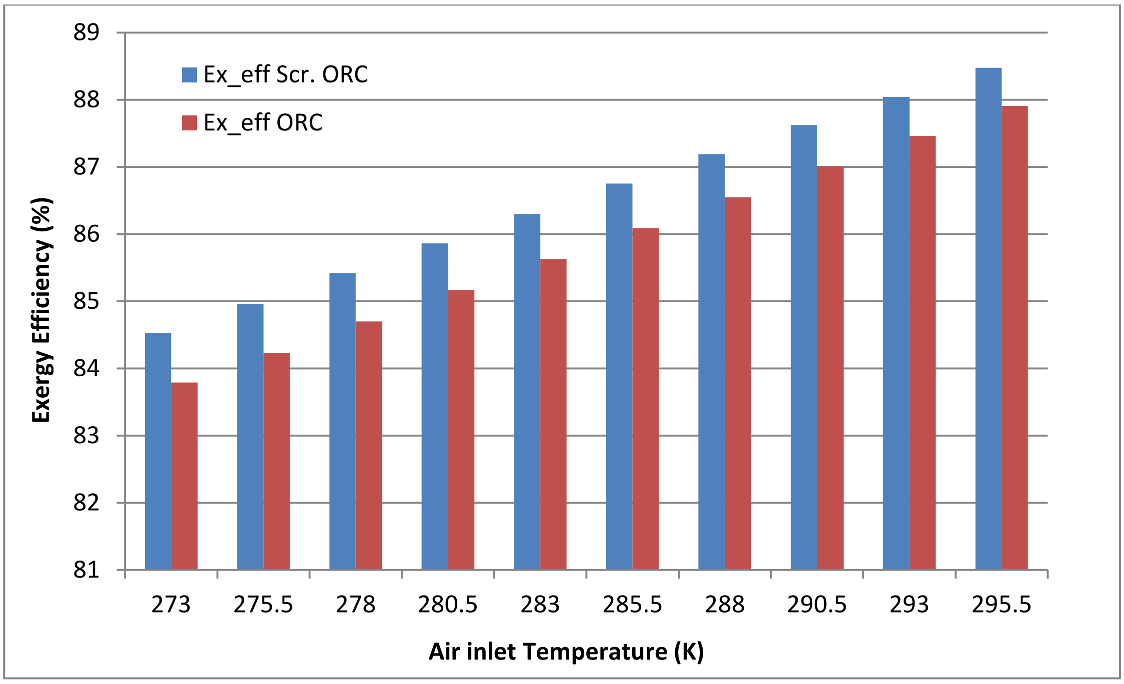


Figure 5.25: Variation of Exergy Efficiency with change in air inlet temperature

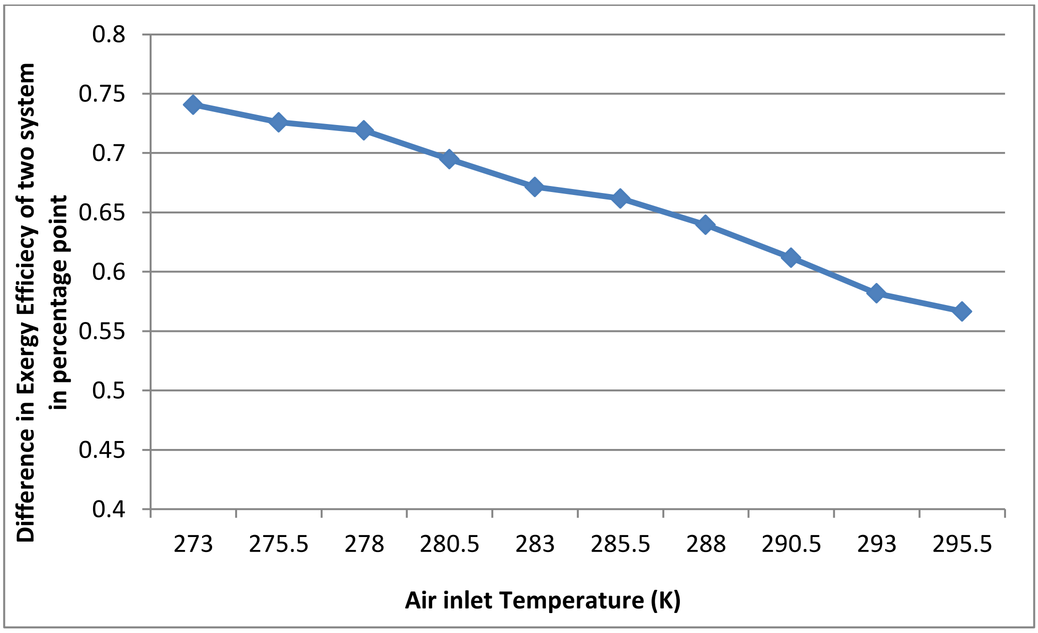


Figure 5.26: Variation of difference in the exergy efficiency with air inlet temperature

Form the two graphs it can be concluded that the system with supercritical ORC as bottoming cycle performs better than system having simple ORC with IHE. However this difference in the performance is more profound in lower atmospheric temperature in compared to the higher air inlet temperature. This makes the system with supercritical ORC to be more favourable for operation in the places where air inlet temperature is comparatively lower. It can be noted that the best performance by the supercritical cycle is obtained at 273 K at which it performs 0.75 percentage point better than the system having simple ORC with IHE as bottoming cycle for the given range of temperature.

5.4.3 Variation of exergy destruction ratio with change in air inlet temperature

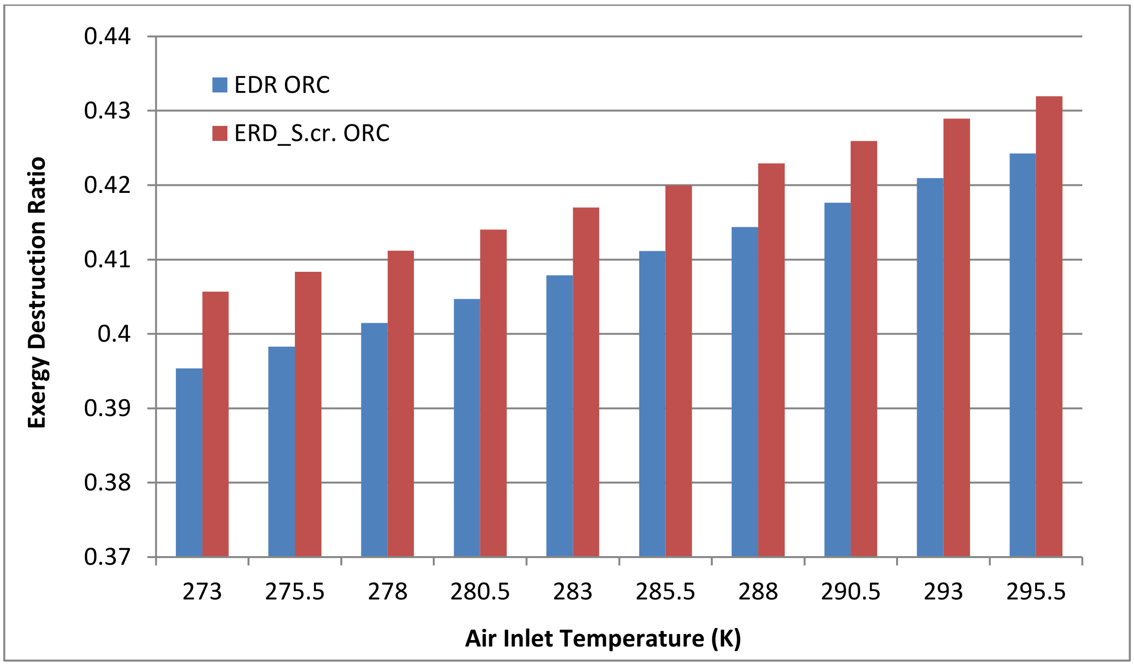


Figure 5.27: Variation of exergy destruction ratio with change in air inlet temperature

Form figure 5.27 it can be observed that with increment in the air inlet temperature the exergy destruction of both the system goes on increasing. The reason for the same can be assigned to the increased in the exergy destruction ay combustion chamber, which act as the primary point of exergy destruction. With increment in the inlet air temperature the mass flow rate of air also increases due to increased work input in case of air compressor. This increase in the air flow causes the increment in the mass flow rate of fuel. This leads to increment in the total exergy destruction taking place at combustion chamber. Hence, the exergy destruction ratio increases with increment in the atmospheric temperature. Now, comparative study of the two system shows that, the exergy destruction obtained in case of supercritical ORC as bottoming cycle is always higher that the exergy destruction obtained in case of system having simple ORC with

IHE as bottoming cycle. However this difference in the exergy destruction ratio goes in decrasing with increase in the air inlet temperature. This can be better explained though the analysis of exergy destruction of the two bottoming cycle individually, apart from rest of the system. Figure 5.28 depicts the change of the exergy destruction of the two bottoming cycle with change in air inlet temperature.

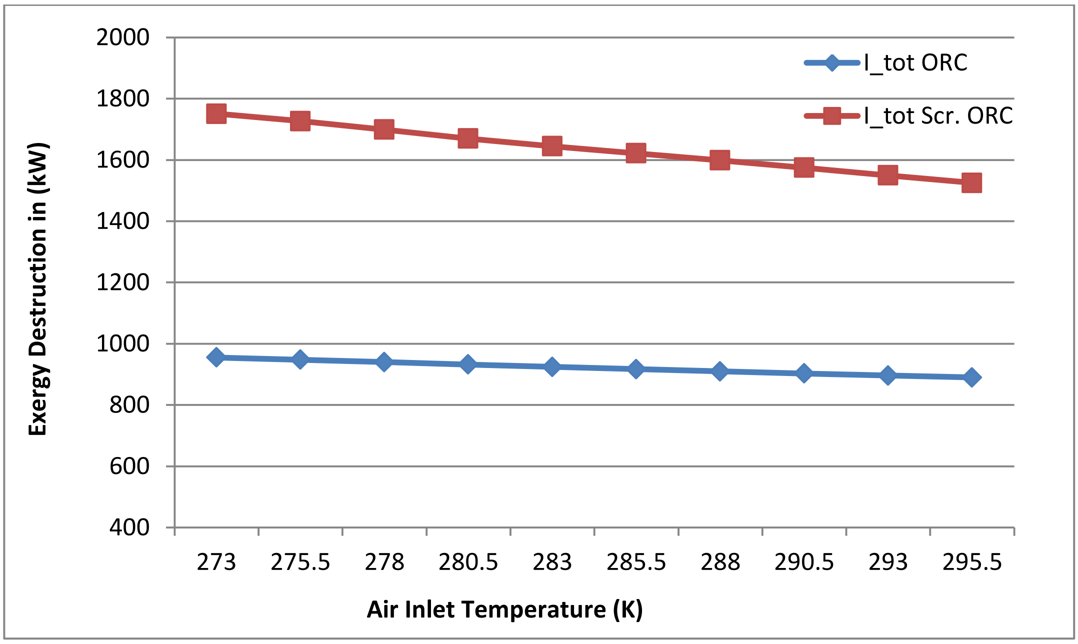


Figure 5.28: Variation of the exergy destruction of the bottoming cycle with air inlet temperature

It can be noted in the diagram that throughout the range of air inlet temperature the exergy destruction in case of supercritical ORC is considerably higher than the simple ORC with IHE. So the difference in the exergy destruction of the two systems can be justified through this variation. Also, the exergy destruction of either of the system decreases with increment in the air inlet temperature. But it can observe that the relative rates of decrease of the two systems are different. The supercritical ORC has a greater rate of decrease in contrast to the simple ORC with IHE. These results reflect on the complete exergy destruction of both the system, where we can observe that the difference between the exergy destruction of the two system narrows down as the air inlet temperature goes on increasing.

5.4.4 Variation of work output of bottoming cycle with air inlet temperature

The work output of the two systems goes on decreasing with increment in the air inlet temperature. The cause for the same can be attributed to increased output at HRSG system which provide with relatively lesser heat to be available for heat recovery at the bottoming cycle. Figure 5.29 depicts the change of the work output of two bottoming cycles with change in air inlet temperature.

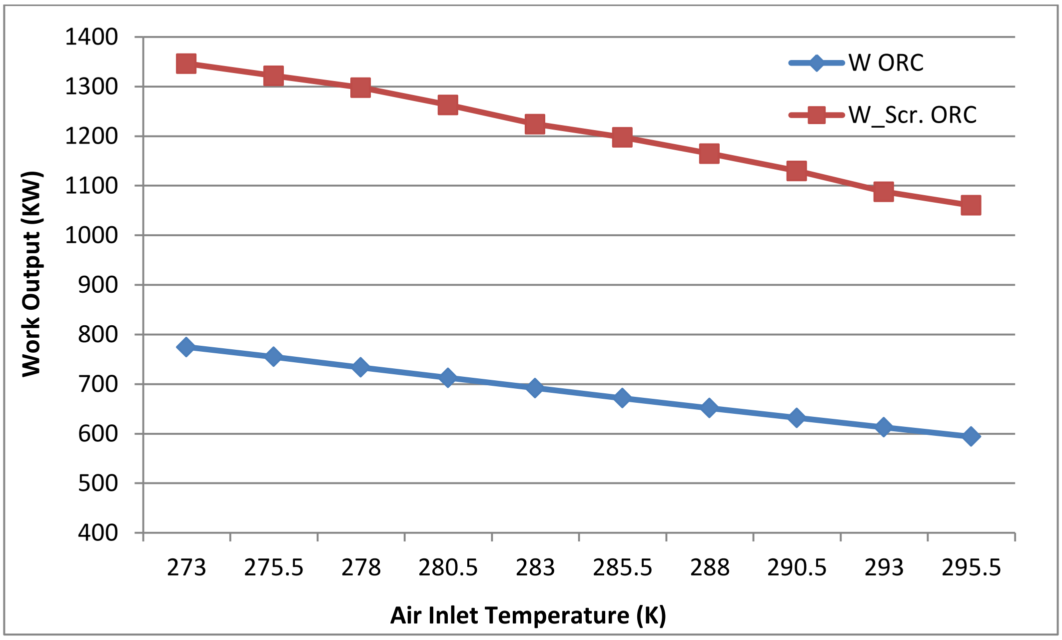


Figure 5.29: Variation of work output of bottoming cycle with air inlet temperature

The comparative study of the bottoming cycle shows that supercritical ORC perform considerably better than the simple ORC with IHE throughout the range of air inlet temperature. However this difference in the work output using the exhaust heat is more profound at lower air inlet temperature. At lower temperature the difference could be as high as the 750 kW. However as the temperature increases this difference goes on decreasing. As it can be seen from the graph that, the rate of decrease is higher in case of supercritical cycle in compared to the rate of decrease of simple ORC with IHE. These results are reflected well in case of first law and exergy examination of the system where it was found that at lower temperature the system with supercritical ORC as the bottoming cycle deliver better performance than the system having simple ORC with IHE and as the air inlet temperature goes on increasing this difference goes on decreasing.

5.5 VARIATION WITH GAS TURBINE ISENTROPIC EFFICIENCY

5.5.1 Variation of first law efficiency with turbine isentropic efficiency

As it can be noted that as there Is increment in isentropic efficiency, the overall efficiency goes on decreasing. The reason for the same can be attributed to decreasing mass flow rate of air. As there is increment in the efficiency of gas turbine, the work output of the gas turbine in the system goes on increasing as due to which the required work output from gas turbine unit become attainable at lower mass flow rate. As the mass flow rate of the system decreases the quantum of energy available to be extracted in the form of process heat also goes on decreasing. This creates an adverse influence on the overall efficiency of the system. As a result it goes on decreasing for both the systems as shown in figure 5.30.

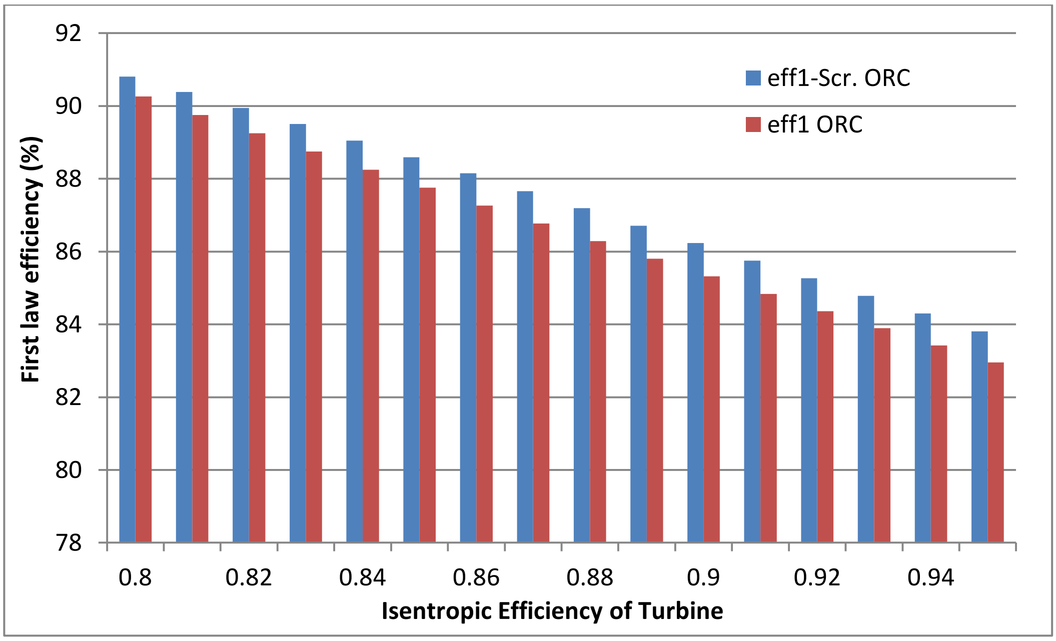


Figure 5.30: Variation of first law efficiency with isentropic efficiency of gas turbine

The comparative study between the systems shows that system with supercritical ORC as bottoming cycle performs better than the system having simple ORC with IHE. The rate of decease of the two systems also comes out to be different. The rate of decrease of first law efficiency in case of system having supercritical ORC as bottoming cycle is comparatively lower than the system having simple ORC with IHE as bottoming cycle.

As a result of this variation the difference in the first law efficiency of the two system goes on decreasing. This variation can also be confirmed though plot in figure 5.31. As it can be concluded that the difference in the first law efficiency of both the system goes on increasing with the isentropic efficiency of the system. It can also be deduced form the figure that the rate of increase in the difference between the efficiency is higher at lower values of isentropic efficiency and as the efficiency reaches its maximum , the rate of increase in the difference also cease to increase.

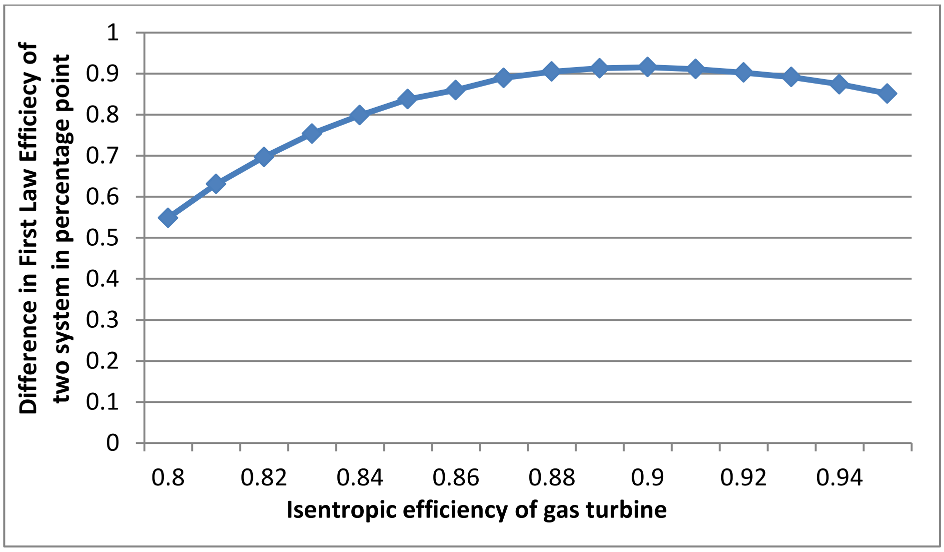


Figure 5.31: Variation of difference in first law efficiency with isentropic efficiency of gas turbine

5.5.2 Variation of exergy efficiency with change in turbine isentropic efficiency

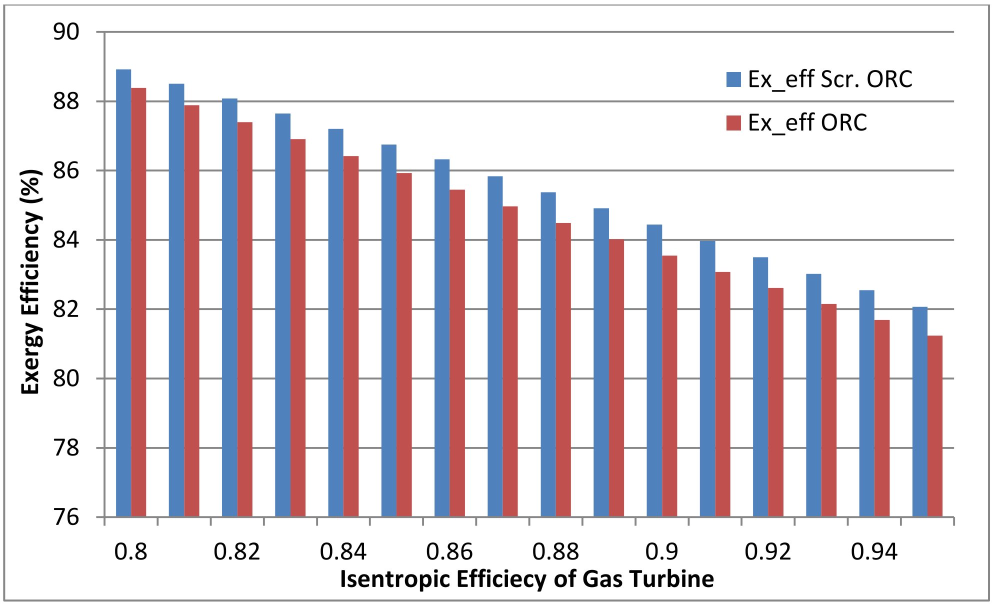


Figure 5.32 : Variation if exergy efficiency with isentropic efficiency of gas turbine

The variation of exergy efficiency is very similar to that of change in first law efficiency. The exergy efficiency of both the system goes on decreasing with increment in the gas turbine isentropic efficiency and there is difference in the rate of decrease. The system with supercritical ORC as bottoming cycle has a lower rate of decrease in compared to the system having simple ORC. This causes the difference between exergy efficiency of the two systems goes on increasing with increment in the gas turbine isentropic efficiency . This result is shown in the figure 5.33 below.

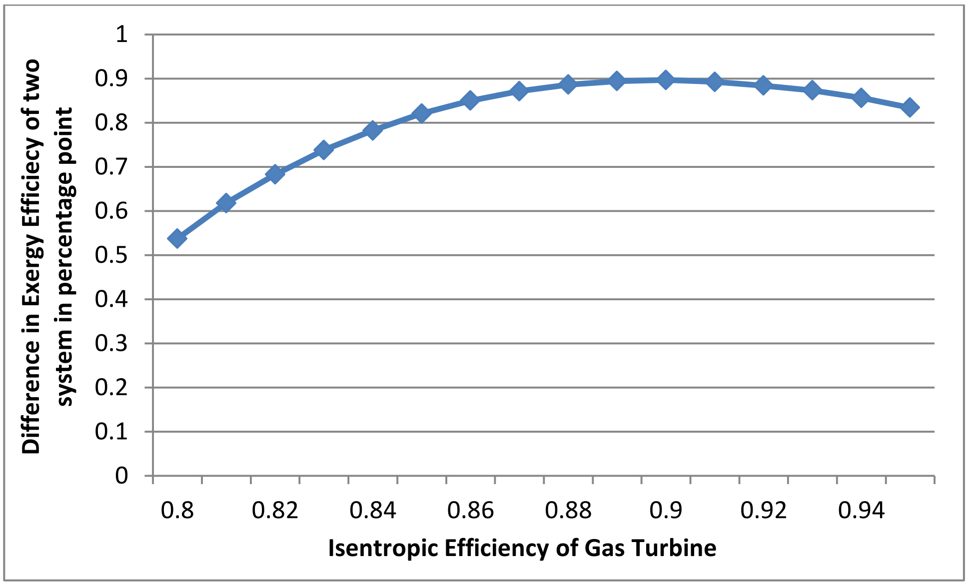


Figure 5.33: Variation of difference in exergy efficiency with isentropic efficiency of gas turbine

5.5.3 Variation of exergy destruction ratio with change in turbine isentropic efficiency

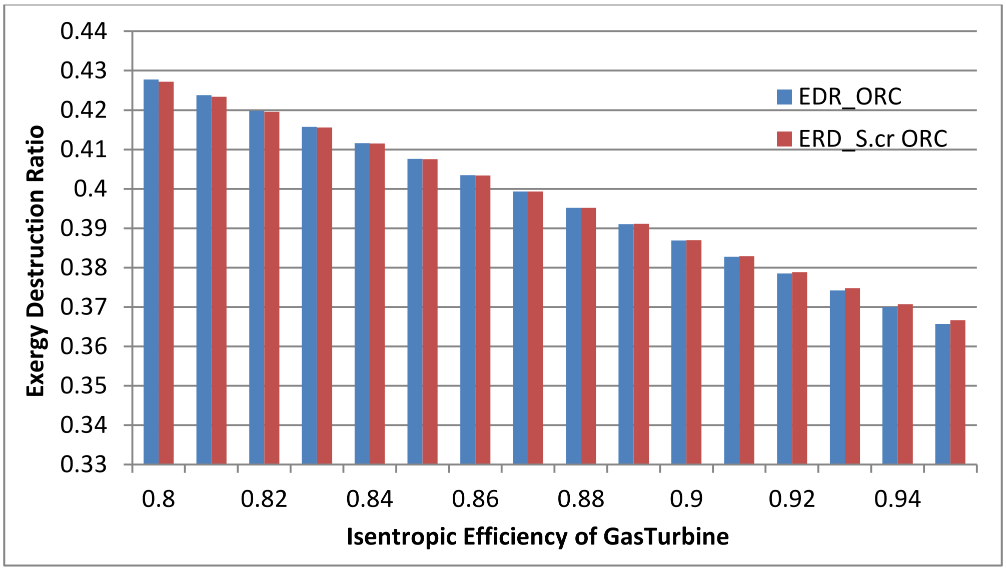


Figure 5.34: Variation of exergy destruction ratio with isentropic efficiency of gas turbine

From figure 5.34 it can be noted that for both the system the exergy destruction ratio goes on decreasing with increment of isentropic efficiency of the system. The reason can be attributed to the decrease in the exergy destruction in combustion chamber, which is a result of decreased fuel supply, which is caused due to decreased mass flow rate of the air. However in contrast to other parametric analysis done for the system, in this parametric study it can be deduced that the two system have equal exergy destruction ratio for almost the entire range of the gas turbine isentropic efficiency. This result can further be validated through study of exergy destruction of individual bottoming cycle apart from the taking system as a whole. Figure 5.35 provides for the variation of the exergy destruction of the two bottoming cycle with change in isentropic efficiency of the system.

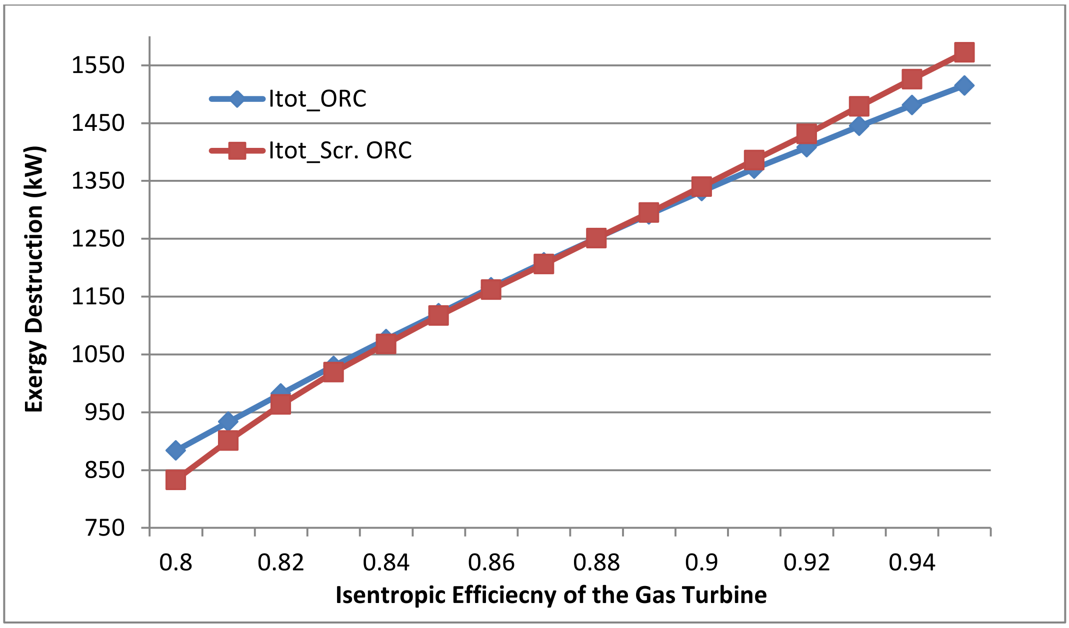


Figure 5.35: Variation of exergy destruction of bottoming cycle with isentropic efficiency of gas turbine

The variation of the exergy destruction of the bottoming cycle is in complete consonance with the graph of exergy destruction ratio. For almost entire range of the isentropic turbine efficiency the exergy destruction obtained from both ths system remains equal. Only slight deviation is obtained at efficiency near to 80% where the exergy destruction of simple ORC is higher in compared to the supercritical ORC as well as the slight variation also occurs at higher value of efficiency i.e., 92% onward after which the exexgy destruction of supercritical ORC overshoots the exergy destruction of the simple ORC with IHE. The result show that the with respect to change in isentropic efficiency the change in the bottoming cycle is of irrelevance in nature as it does not creates a significant difference throughout the range.

5.5.4 Variation of work output of bottoming cycle with change in turbine isentropic efficiency

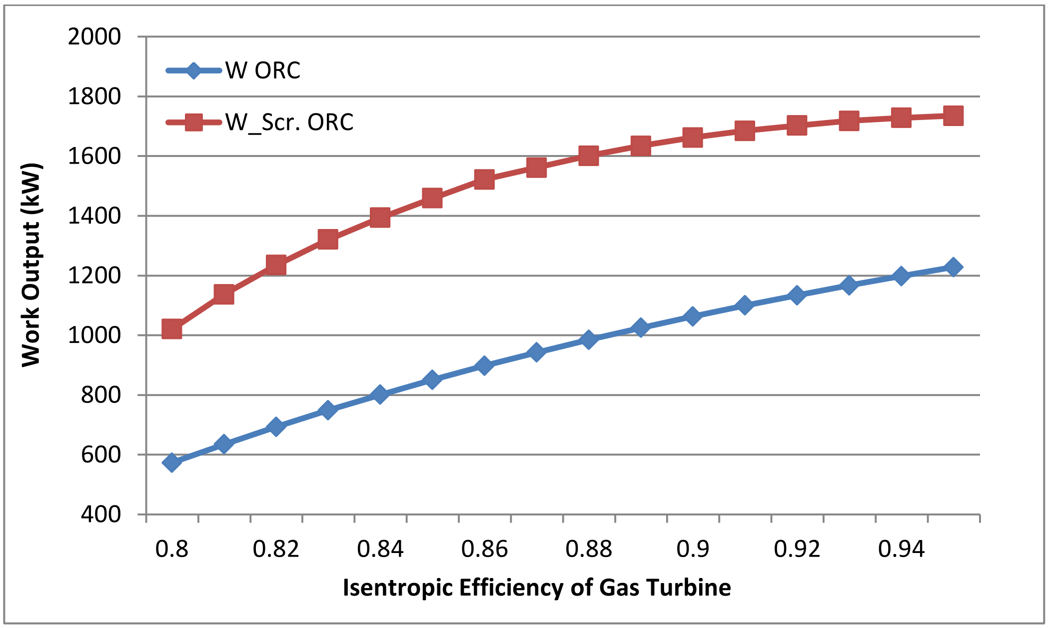


Figure 5.36: Variation of work output of bottoming cycle with isentropic efficiency of gas turbine

With increase in the gas turbine isentropic efficiency, the work output of both the bottoming cycle goes on increasing. The reason for increase in the work output can be ascribed to poor performance of HRSG system due to notable decrement in mass flow rate of air. This leaves the air coming out as exhaust from HRSG system with greater heat energy which can be recovered easily in the bottoming cycle. The comparative study of the system show that amount of heat recovered by the supercritical ORC system remains significantly higher than the simple ORC system with IHE. While simple ORC system shows a linear rate of increase in work output, the supercritical ORC has a greater rate of increase at lower isentropic efficiency and he rate of increase decreases with increment in the isentropic efficiency. Further it can be deduced that the increased work output of the bottoming cycle is unable to compensate the decreased output at HRSG system, which has a dominating influence on the overall system and hence the overall first law efficiency of the system goes on decreasing with increment in gas turbine isentropic efficiency, as obtained in the results of preceding sections.

CHAPTER 6

CONCLUSION

In this work the examination of two cogeneration systems is carried out to provide a comparative study between them. These system included GT-HRSG-ORC cogeneration system and GT-HRSG-Supercritical ORC. The objective of the work is to analyse if the use of supercritical ORC in the bottoming cycle can help in enhancing the overall performance of the system. The parametric analysis of the system is performed to understand the relative change in behaviour of the two systems. Based on the relative study of the two system following conclusions are drawn.

- Parametric study of the two systems shows that, with variation of the concerned parameter the change in behaviour of both the system is similar. Although the rate at which two system vary with change in a particular parameter can be different from one another.
- The two component at the bottoming side of gas turbine i.e., HRSG system and ORC system act in complementary manner to one another. Rather than two system simultaneously enhancing the performance of overall system. The improvement in the output of one system can cause the decrease in the output of other. So loss in one system is compensated by other and this is true irrespective of which ORC system is used in bottoming cycle.
- The HRSG system is highly responsive to change in output of the exhaust from the APH system along with the flow rate of the exhaust gasses. The decrease in either one of them creates an impact on the performance of the HRSG system in negative way.
- HRSG system has a dominating influence in the overall system efficiency because decrease in the output of HRSG always have negative impact on overall performance of the system and both supercritical as well as simple ORC system, though acting as a complementary system to recover waste energy, are unable to

negate the losses that occur due to loss in performance of HRSG system due to their comparatively low work output.

- Comparative study between the two system establish that system with supercritical ORC performs better than the system having subcritical ORC system in the bottoming cycle during energy and exergy analysis of the system irrespective the parameter used in the study.
- For both energy and exergy analysis the system with supercritical ORC register an increase in efficiency in compared to system with subcritical ORC, ranging from 0.6 percentage point to 1.2 percentage point depending on the parametric study carried out for the systems.
- In case of the analysis carried out to calculate the EDR for the system. It was observed that the system shows variable results depending upon the parameter used in the study.
- Comparative study of work output of the bottoming cycle establish that the work output obtained from supercritical ORC remains significantly higher than the work output of the subcritical ORC. As a result the supercritical ORC provide better heat recovery from exhaust gasses.
- It can be also be concluded that for obtaining a given performance of the system, the supercritical ORC cycle in bottoming cycle reduce the constrain that are associated with designing of the gas turbine system in topping cycle. It can help in providing same performance of the system at lower TIT as well as at lower compression ratio improving both, the economics as well as durability of the system.
- The system with supercritical ORC possesses a major challenge in terms of EDR which remain quite higher than system with subcritical ORC. It can also have a unfavourable impact on the environment as it reduces the exhaust gas final temperature below 400 K which is a challenging aspect and need to be addressed.

Supercritical ORC can be a good replacement to the subcritical ORC cycle in the GT-HRSG cogeneration for enhancing the performance of the system with regard to both energy and exergy, provided the concerns regarding the environmental issue associated with the system are addressed properly.

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