THERMODYNAMIC EVALUATION OF BRAYTON-RANKINE COMBINED CYCLE WITH REHEATING

A project report submitted in partial fulfilment of the requirements for the award of the degree of

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

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We are happy to present this report on “THERMODYNAMIC EVALUATION OF BRAYTON-RANKINE COMBINED CYCLE WITH REHEATING” in partial fulfilment of the requirement for the award of B.E., Degree in Mechanical Engineering.

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ABSTRACT

Energy and exergy analysis is a strong tool in assessing the functioning of power plants, process plants, refineries etc. The true losses in thermodynamic systems can be best evaluated based on second law or exergy analysis. The identification of irreversibilities in a cyclic process and reducing them will lead to maximum utilization of energy. This will have a positive impact on the use of natural resources and environmental pollution by reducing the rate of waste heat.

The present work involves thermodynamic evaluation of a combined cycle power plant which adopts reheating in the bottoming cycle. It consists of evaluating the first and second law efficiencies, generally termed as energy and exergy analysis of the combined Brayton-Rankine power cycle system. Further a parametric analysis is also performed to assess the effect of important variables like pressure ratio, the peak temperature, steam turbine inlet pressure, condenser pressure, heat recovery steam generator efficiency , reheat pressure .The energy efficiency, exergy efficiency, power ratio and exergy destruction are evaluated for study. It is observed that with increase in maximum temperature and pressure ratio of the gas cycle, the efficiencies perk up. It is observed that the power outputs of gas and steam cycles are strongly dependent on the pressure ratio of the topping cycle. The power ratio which is here defined as the ratio of power outputs of gas to the steam cycles is almost one for pressure ratios in the range of 3 to 3.5.The exergy destruction is found to increase at higher peak temperatures but declines with enhancing pressure ratios. The turbine efficiency when chosen as a parameter exhibits optimum pressure ratio for the maximum efficiency values. Similar results were also obtained, when compressor efficiency was taken up for parametric analysis.

Heat recovery steam generator plays a pivotal role in transferring heat from gas cycle to steam cycle. Its first law efficiency when enhanced produces a positive impact on the performance of the plant. The effect of maximum steam pressure is studied in conjunction with the pressure ratio of gas cycle. It is observed that at lower pressure ratios of gas cycle, highest steam pressure produce higher energy efficiency and vice-versa. A similar trend is also observed with respect to exergy efficiency. The influence of condenser pressure is higher at low pressure ratios and diminishes gradually at higher pressure ratio values.

The highest exergy destruction was observed to be occurring in the combustion chamber amounting to about 67%. The second law efficiencies of individual equipments were also evaluated. The turbines exhibit the highest second law efficiencies while the feed pump has the lowest value.

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NOMENCLATURE

η	Efficiency
C_p	Specific heat at constant pressure (kJ/kg-K)
C_v	calorific value of fuel (kJ/kg)
e	specific energy (kJ/kg)
h	Specific enthalpy (kJ/kg)
m_f	Mass flow rate of fuel (kg/s)
m_g	Mass flow rate of gas (kg/s)
m_s	Mass flow rate of steam (kg/s)
P	Pressure (bar)
Q_s	Heat supplied (kJ)
P_{net1}	Net power output of gas turbine (kW)
P_{net2}	Net power output of steam turbine (kW)
T_0	Dead state temperature (K)
T_{sat}	Saturation temperature (K)

v	Specific volume (m^3/kg)
x	Dryness fraction
ED	Exergy destruction (kW)

Subscripts

f	Saturation liquid state
g	Saturation
I	First law
II	Second law
i	Inlet
e	Exhaust
c	Condenser
fp	Feed pump
rh	Reheater
hrsg	Heat recovery steam generator

CONTENTS

CHAPTER 1

1 INTRODUCTION

Thermodynamics is the science that deals with heat and work and those properties of substances that bear a relation to heat and work. Like all sciences, the basis of thermodynamics is experimental observation. In thermodynamics these findings have been formalized into certain basic laws, which are known as the first, second, and third law of thermodynamics. In addition to these laws, the zeroth law of thermodynamics, which in the logical development of thermodynamics precedes the first law, has been set forth.

We can redefine thermodynamics as the science of energy &exergy referred to as the first law and second law of thermodynamics. This will make it thermodynamically more correct because both energy and exergy quantities possess the same units and the efficiency can be defined from both energy and exergy point of views. Consequently, exergy analysis can assist in improving and optimizing designs and analysis. Two key features of exergy analysis are

1. It yields efficiencies that provide a true measure of how nearly actual performance approaches the ideal, and
2. It identifies more clearly than energy analysis the types, causes and locations of thermodynamics losses.

1.1 ENERGY

Energy is the property that must be transferred to a system in order to perform work on, or to heat, the system. Energy is a conserved quantity; the law of conservation of energy states that energy can be converted in form, but not created or destroyed. The SI unit of energy is the Joule, which is the energy transferred to a system by the work of moving it a distance of 1 metre against a force of 1 newton.

Common forms of energy include the kinetic energy of a moving system, the potential energy stored by an system's position in a force field (gravitational, electric or magnetic), the elastic energy stored by stretching solid systems, the chemical energy released when a fuel burns, the radiant energy carried by light, and the thermal energy due to an system's temperature.

Energy exists in numerous forms such as thermal, mechanical, electric, chemical, and nuclear. Even mass can be considered a form of energy. Energy can be transferred to or from a closed system (a fixed mass) in two distinct forms: heat and work. For control volumes, energy can also be transferred by mass flow. An energy transfer to or from a closed system is heat if it is caused by a temperature difference. Otherwise it is work, and it is caused by a force acting through a distance.

Thermodynamics provides no information about the absolute value of the total energy. It deals only with the change of the total energy, which is what matters in engineering problems. Thus the total energy of a system can be assigned a value of zero at some convenient reference point. The change in total energy of a system is independent of the reference point selected. The decrease in the potential energy of a falling rock, for example, depends on only the elevation difference and not the reference level selected. In thermodynamic analysis, it is often helpful to consider the various forms of energy that make up the total energy of a system in two groups: macroscopic and microscopic. The macroscopic forms of energy are those a system possesses as a whole with respect to some outside reference frame, such as kinetic and potential energies. The microscopic forms of energy are those related to the molecular structure of a system and the degree of the molecular activity, and they are independent of outside reference frames. The sum of all the microscopic forms of energy is called the internal energy of a system and is denoted by U. The term energy was coined in 1807 by Thomas Young, and its use in thermodynamics was proposed in 1852 by Lord Kelvin. The term internal energy and its symbol U first appeared in the works of Rudolph Clausius and William Rankine in the second half of the nineteenth century, and it eventually replaced the alternative terms inner work, internal work, and intrinsic energy commonly used at the time. The macroscopic energy of a system is related to motion and the influence of some external effects such as gravity, magnetism, electricity, and surface tension. The energy that a system possesses as a result of its motion relative to some reference frame is called kinetic energy (KE).

1.2 EXERGY

The term Exergy was used for the first time by Rant in 1956, and refers to the Greek words ex (external) and ergos (work). Another term describing the same is Available Energy or simply Availability. The term Exergy also relates to ideal work and exergy Losses relate to Lost Work. Exergy can also be described in a similar but somewhat more complicated way.

Szargut' used the following statement to explain the term:

"Exergy is the amount of work obtainable when some matter is brought to a state of thermodynamic equilibrium with the common components of its surrounding nature by means of reversible processes, involving interaction only with the above mentioned components of nature".

Energy =Exergy+Anergy

1.3 EXERGY ANALYSIS

An exergy analysis identifies the location, the magnitude and the sources of thermodynamic inefficiencies in a thermal system. This information, which cannot be provided by other means (e.g., an energy analysis), is very useful for improving the overall efficiency and cost effectiveness of a system or for comparing the performance of two systems.

Exergy can be regarded as a measure of the usefulness or quality of energy. Technically, exergy is defined as the maximum amount of work that can be produced by a stream of energy or matter, or from a system, as it is brought into equilibrium with a reference environment. Unlike energy, exergy is consumed during real processes due to irreversibility's and conserved during ideal processes. Exergy and related concepts have been recognized for more than a century.

Exergy analysis is a methodology that uses the first and second laws of thermodynamics for the analysis, design, and improvement of energy and other systems. The exergy method is useful for improving the efficiency of energy-resource use, for it quantifies the locations, types, and magnitudes of wastes and losses. In general, more meaningful efficiencies are evaluated with exergy analysis rather than energy analysis because exergy efficiencies are always a measure of the approach to the ideal. Therefore, exergy analysis accurately identifies the margin available to design more efficient energy systems by reducing inefficiency

1.4 FIRST LAW OF THERMODYNAMICS

The First Law of thermodynamics may be stated in several ways : The increase in internal energy of a closed system is equal to total of the energy added to the system. In particular, if the energy entering the system is supplied as heat and if energy leaves the system as work, the heat is accounted for as positive and the work as negative

$\Delta U = Q - W$

In the case of a thermodynamic cycle of a closed system, which returns to its original state, the heat Q_{in} supplied to the system in one stage of the cycle, minus the heat Q_{out} removed from it in another stage of the cycle, plus the work added to the system W_{in} equals the work that leaves the system W_{out}.

More specifically, the First Law encompasses several principles:

The law of conservation of energy

This states that energy can be neither created nor destroyed. However, energy can change forms, and energy can flow from one place to another. A particular consequence of the law of conservation of energy is that the total energy of an isolated system does not change.

1.5 SECOND LAW OF THERMODYNAMICS

The second law of thermodynamics indicates the irreversibility of natural processes, and, in many cases, the tendency of natural processes to lead towards spatial homogeneity of matter and energy, and especially of temperature. It can be formulated in a variety of interesting and important ways.

It implies the existence of a quantity called the entropy of a thermodynamic system. In terms of this quantity it implies that The second law is applicable to a wide variety of processes, reversible and irreversible. All natural processes are irreversible. Reversible processes are a useful and convenient theoretical fiction, but do not occur in nature.

A prime example of irreversibility is in the transfer of heat by conduction or radiation. It was known long before the discovery of the notion of entropy that when two bodies initially of different temperatures come into thermal connection, then heat always flows from the hotter body to the colder one.

The second law tells also about kinds of irreversibility other than heat transfer, for example those of friction and viscosity, and those of chemical reactions. The notion of entropy is needed to provide that wider scope of the law.

According to the second law of thermodynamics, in a theoretical and fictive reversible heat transfer, an element of heat transferred, δQ, is the product of the temperature (T), both of the system and of the sources or destination of the heat, with the increment (dS) of the system's conjugate variable, its entropy (S)

$$dQ = Tds$$

1.6 ENERGY BALANCE FOR STEADY FLOW SYSTEM

Most devices encountered in practice are steady flow devices, such as turbine, heat exchanger, pumps, condensers, etc. Their mass, energy and entropy and volume remain given by the expression.

The steady flow energy equation represents the energy balance in a steady flow process. It is given by the expression,

$$h_1 + \frac{{C_1}^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{{C_2}^2}{2} + gz_2 + W_s$$

Applying the steady flow energy equation to the turbine while neglecting potential energy and kinetic energy and assuming that the process is isentropic then

$$h_1 = h_2 + W_s$$

Similarly by applying steady flow energy equation to a pump while neglecting potential energy and kinetic energy and assuming that the process is isentropic, we have,

$$h_2 = h_1 + W_s$$

$$W_s = h_2 - h_1$$

1.7 SECOND LAW EFFICIENCY (EXERGY EFFICIENCY)

A common measure on energy is the first law efficiency. The first law efficiency is defined as the Ƞconcerned only with the quantities of energy, disregards the forms in which the energy exists. It does not also discriminate between the energies available at different temperatures. It is the second law of thermodynamics which provides a means of assigning a quality index to energy. The concept of available energy or exergy provides a useful measure of energy quality.

With this concept, it is possible to analyse means of minimizing the consumption of available energy to perform a given process, thereby ensuring the most efficient possible conversion of energy from the required task.

The second law efficiency of process is defined as the ratio of minimum available energy or exergy which must be consumed to do a task divided by the actual amount of energy consumed in performing the task.

$$\eta_{II} = \frac{minimum\ exergy\ intake\ to\ perform\ the\ given\ task}{actual\ exergy\ intake\ to\ perform\ the\ same\ task}$$

This definition is specifically used for devices like turbines, pumps, etc. which either produce or consume shaft work.

The second law efficiency can also be defined as the ratio of exergy recovered from a system to the exergy supplied. This definition holds for devices like heat exchangers, boilers, etc.

$$\eta_{II} = \frac{Energy\ recovered}{Energy\ supplied}$$

1.7.1 TURBINE

The second law efficiencies for steady flow devices like turbines, compressors, heat exchangers and mixing chambers can be determined from their definitions. For a work producing devices such as turbine, the second law efficiency is defined as

$$\eta_{II\ turbine} = \left(\frac{W_u}{W_{rev}}\right) = \left(\frac{W_{act}}{W_{max}}\right)$$

Where W_u is the actual useful work and W_{rev} is the reversible work

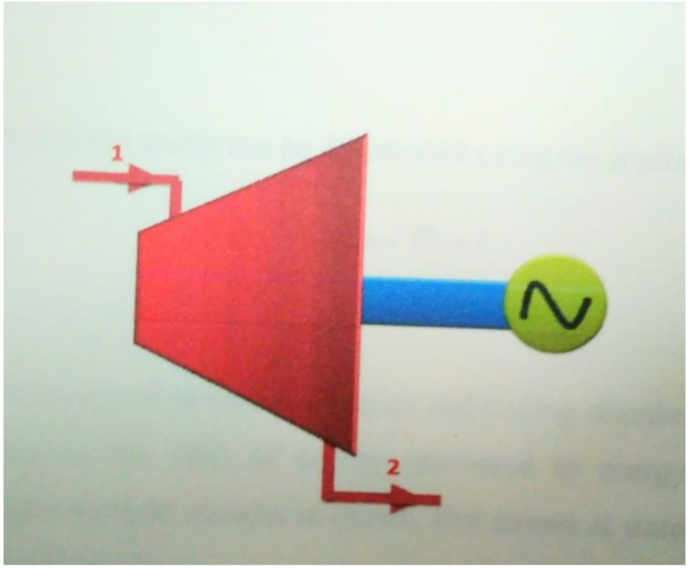


Fig 1.1 Block diagram of a turbine

The actual work of the turbine $W_{act} = h_1 - h_2$

The reversible work can be determined by setting the exergy destructions to zero in the exergy balance relation. For one inlet- one exit steady flow devices, such as turbine, it is

$$W_{max} = (h_i - h_e) - T_0(S_i - S_e)$$

1.7.2 PUMP

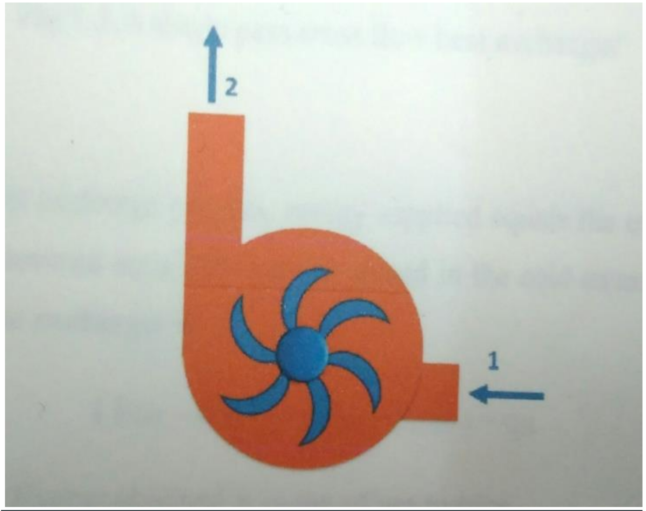


Fig 1.2 Block diagram of a pump

$$W_{act} = h_2 - h_1$$

$$W_{min} = (h_i - h_e) - T_0(S_i - S_e)$$

The second law efficiency of a pump can e determined using a similar method. It is given as

$$\eta_{II\ pump} = \frac{W_{min}}{W_{act}}$$

1.7.3 HEAT EXCHANGER

Since no work is involved in heat exchangers and mixing chambers, their second law efficiencies are defined as the ratio of exergy recovered to exergy supplied. A heat exchanger with two unmixed fluid streams is shown. Hot stream at state-I and state-II, cold stream enters at state 3 and leaves.

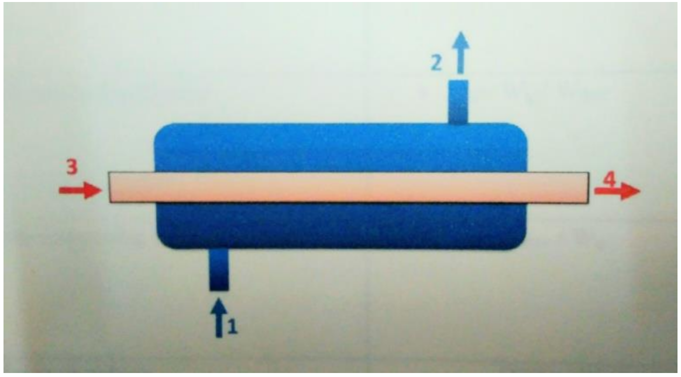


Fig 1.3 A single pass cross flow heat exchanger

During the heat exchange process, exergy supplied equals the exergy lost in the hot stream and exergy recovered equals the exergy gained in the cold stream. Thus, the second law efficiency of a heat exchanger is

$$\eta_{II} = \frac{Exergy\ increase\ of\ cold\ fluid}{Exergy\ decrease\ of\ hot\ fluid}$$

The second law efficiencies of some devices are summarised in the following tab

Table 1.1 Second law efficiencies of various devices

DEVICES	SECOND LAW EFFICIENCY
Heat engine	$\eta_{II} = \frac{\eta_{th}}{\eta_{th.rev}}$

Work producing Devices	$\eta_{II} = \frac{W_u}{W_{Rev}}$
Work consuming devices	$\eta_{II} = \frac{W_{Rev}}{W_u}$
Mixing chambers	$\eta_{II} = \frac{Exergy\ recovered}{Exergy\ supplied}$

CHAPTER 2

2 RANKINE-BRAYTON COMBINED CYCLE

2.1 RANKINE CYCLE:

The Rankine cycle is the fundamental operating cycle of all power plants where an operating fluid is continuously evaporated and condensed. The selection of operating fluid depends mainly on the available temperature range.

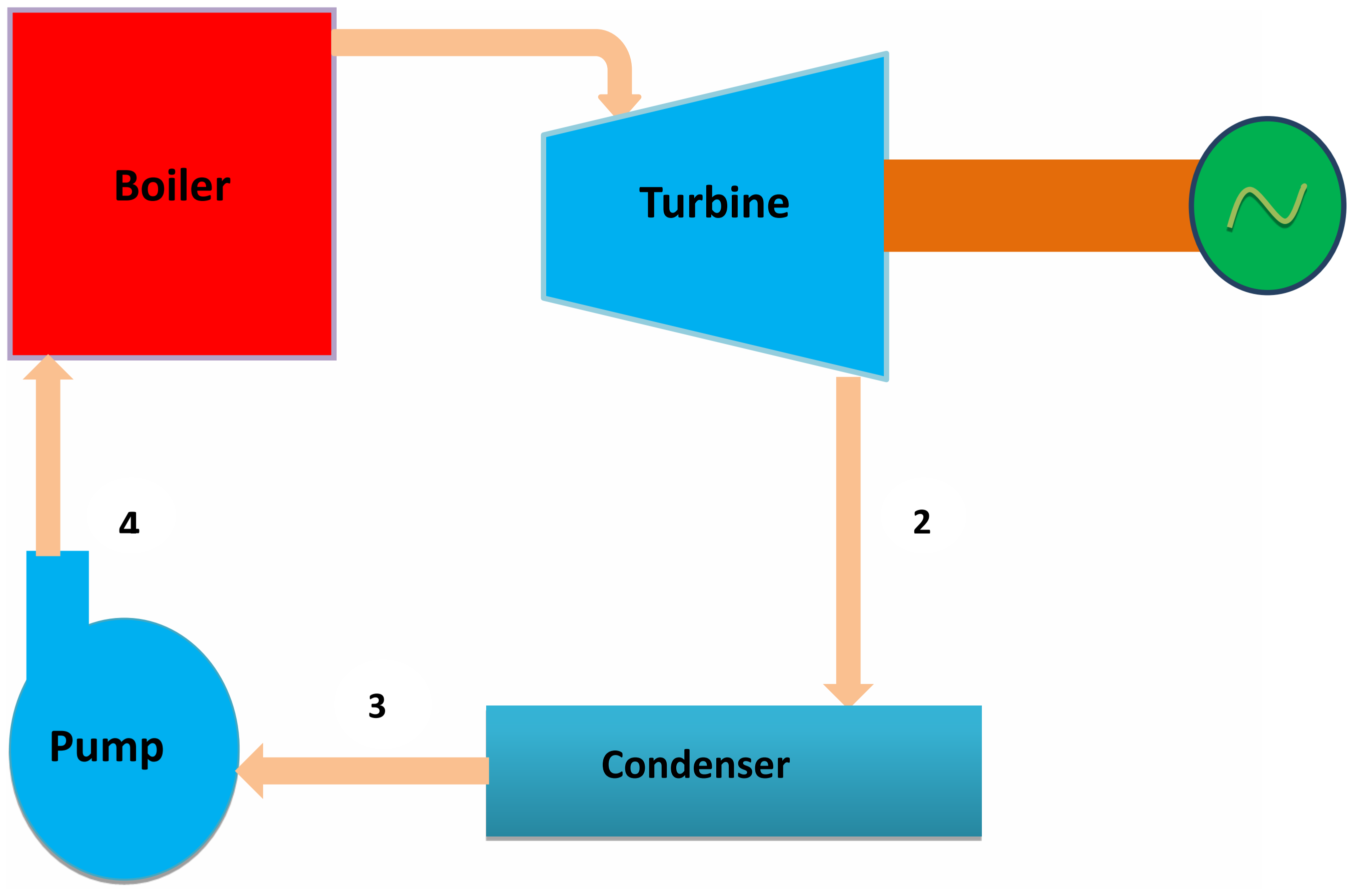
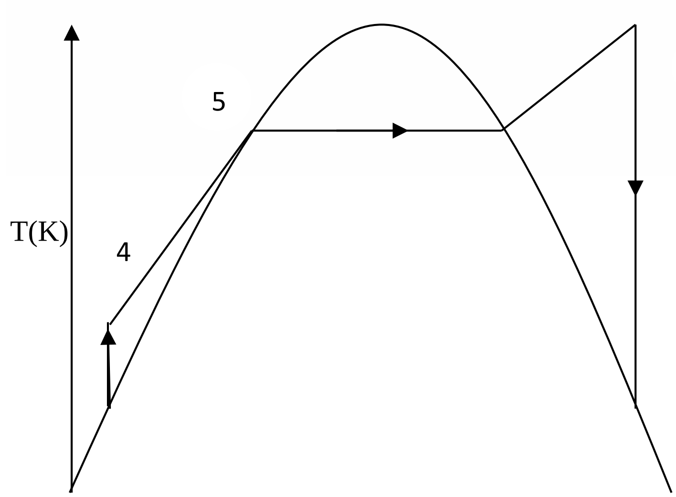


Fig 2.1 Layout of Rankine cycle



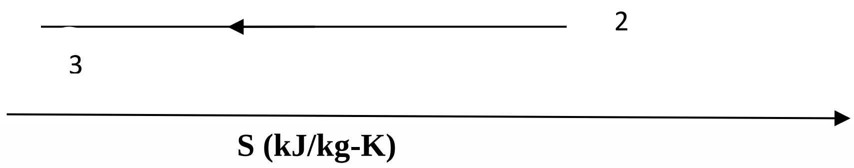


Fig 2.2 T-s plot of a Rankine cycle

The Rankine cycle operates in the following steps:

- 4-1 **Isobaric Heat Transfer**. High pressure liquid enters the boiler from the feed pump and is heated to the saturation temperature. Further addition of energy causes evaporation of the liquid until it is fully converted to saturated steam.
- 1-2 **Isentropic Expansion**. The vapour is expanded in the turbine, thus producing work which may be converted to electricity. In practice, the expansion is limited by the temperature of the cooling medium and by the erosion of the turbine blades by liquid entrainment in the vapour stream as the process moves further into the two-phase region. Exit vapour qualities should be greater than 90%.
- 2-3 **Isobaric Heat Rejection**. The vapour-liquid mixture leaving the turbine is condensed at low pressure, usually in a surface condenser using cooling water. In well designed and maintained condensers, the pressure of the vapour is well below atmospheric pressure, approaching the saturation pressure of the operating fluid at the cooling water temperature.
- 3-4 **Isentropic Compression**. The pressure of the condensate is raised in the feed pump. Because of the low specific volume of liquids, the pump work is relatively small and often neglected in thermodynamic calculations.

2.2 BRAYTON CYCLE:

The Brayton cycle (or Joule cycle) represents the operation of a gas turbine engine. The cycle consists of four processes, as shown in Figure alongside a sketch of an engine:

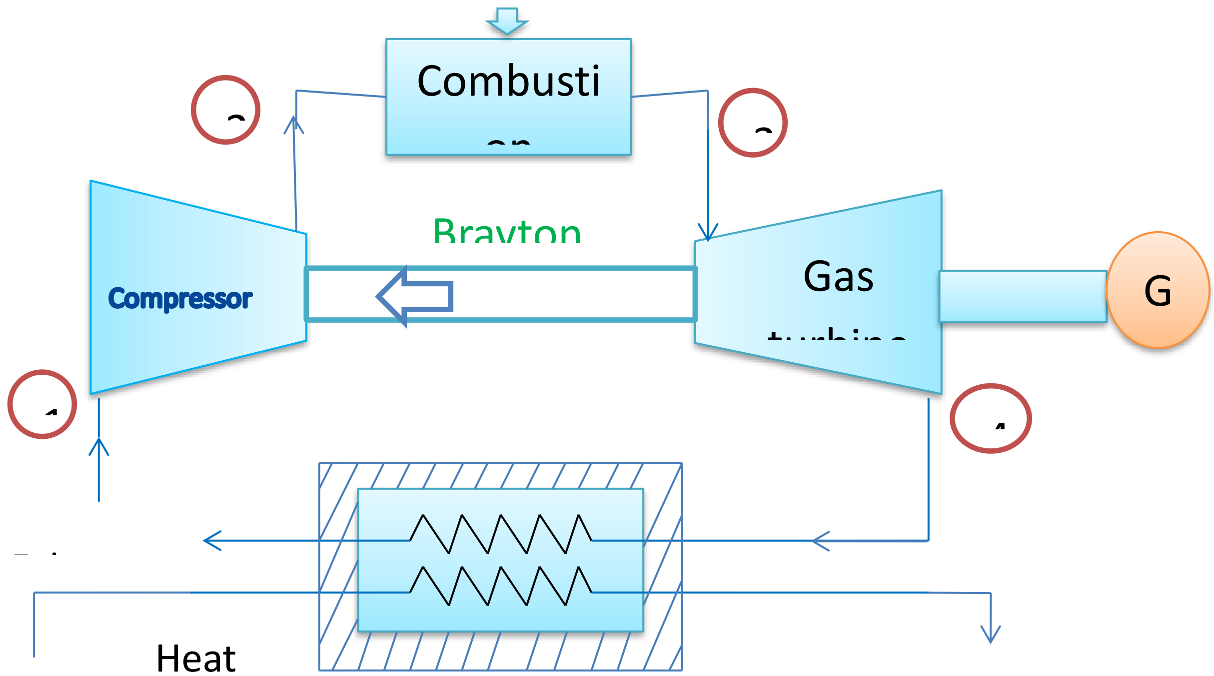
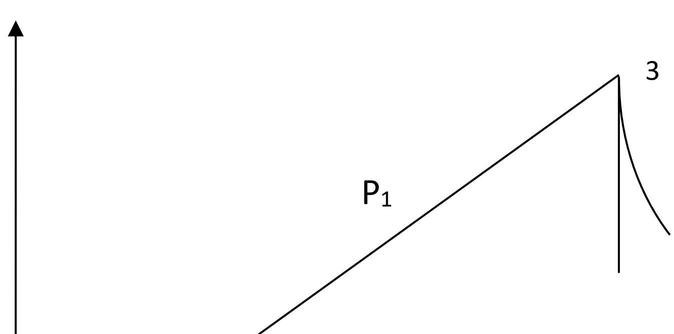


Fig 2.3 Layout of Brayton cycle

- 1- 2 Adiabatic, quasi-static (or reversible) compression in the inlet and compressor;
- 2- 3 Constant pressure fuel combustion (idealized as constant pressure heat addition);
- 3- 4 Adiabatic, quasi-static (or reversible) expansion in the turbine and exhaust nozzle, with which we
    1. take some work out of the air and use it to drive the compressor, and
    2. take the remaining work out and use it to accelerate fluid for jet propulsion, or to turn a generator for electrical power generation;
- 4 - 1 Cool the air at constant pressure back to its initial condition.



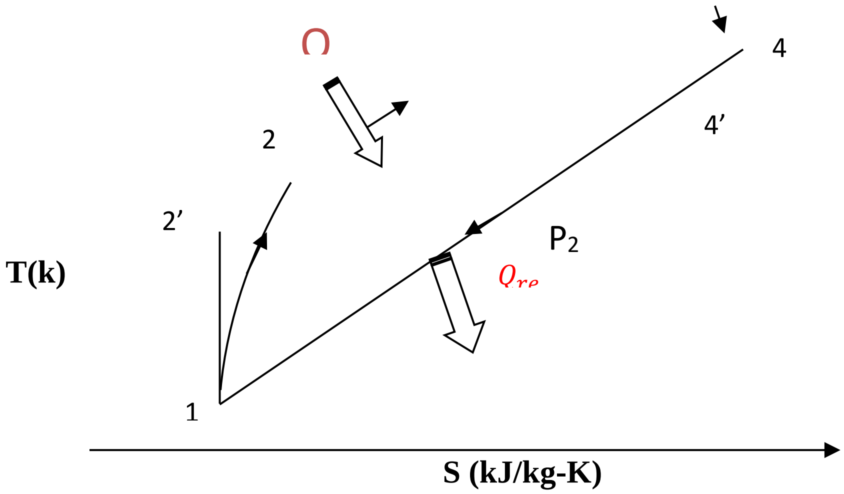


Fig 2.4 T-s plot of Brayton cycle

2.3 COMBINED BRAYTON-RANKINE POWER CYCLE WITH REHEATING :

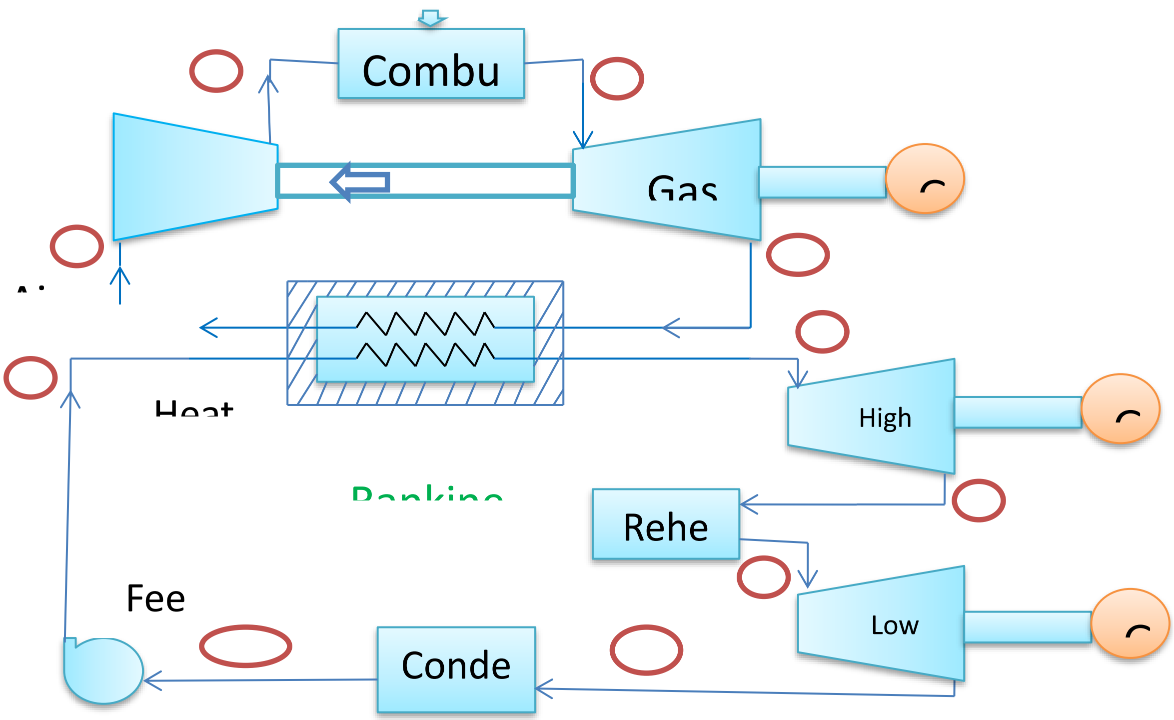


Fig 2.5 Flow diagram of a combined Brayton and Rankine cycle

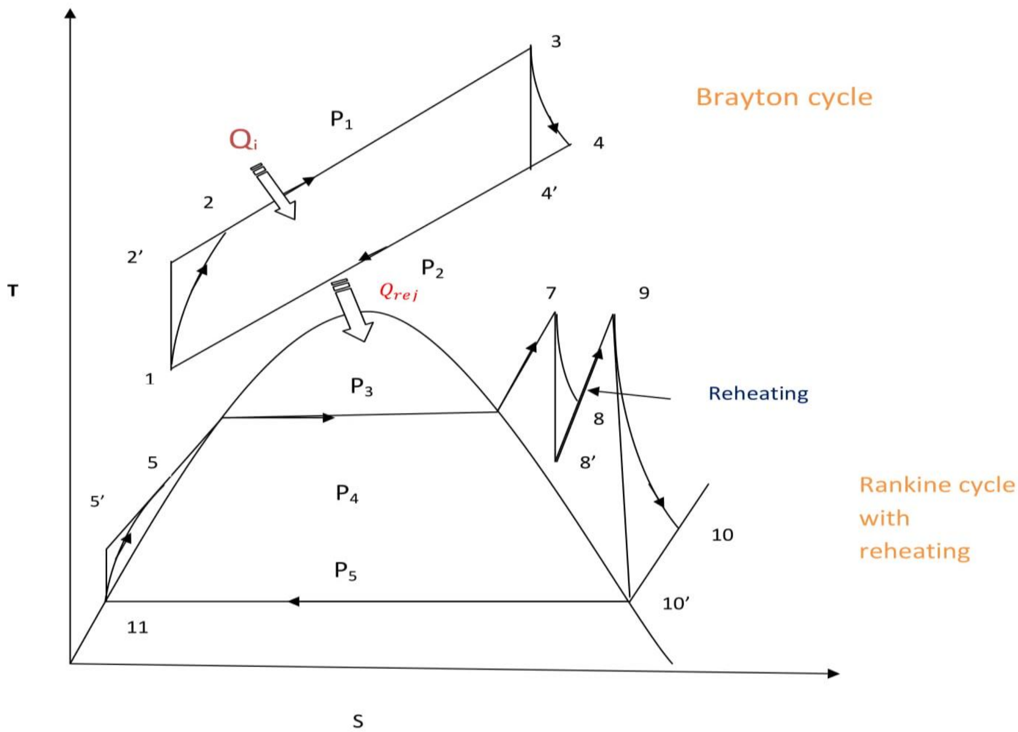


Fig: TS Diagram for combined brayton and rankine cycle

pinch point: It refers to the states of the liquid and gas when their temperature difference is minimum. The temperature difference corresponding to this point is known as pinch point temperature difference. This point has special significance as it is necessary to consider this in the heat balance equations. Generally for thermal design, when one of the fluid is undergoing a phase change, the pinch point temperature difference is assumed. This is as indicated in the figure below

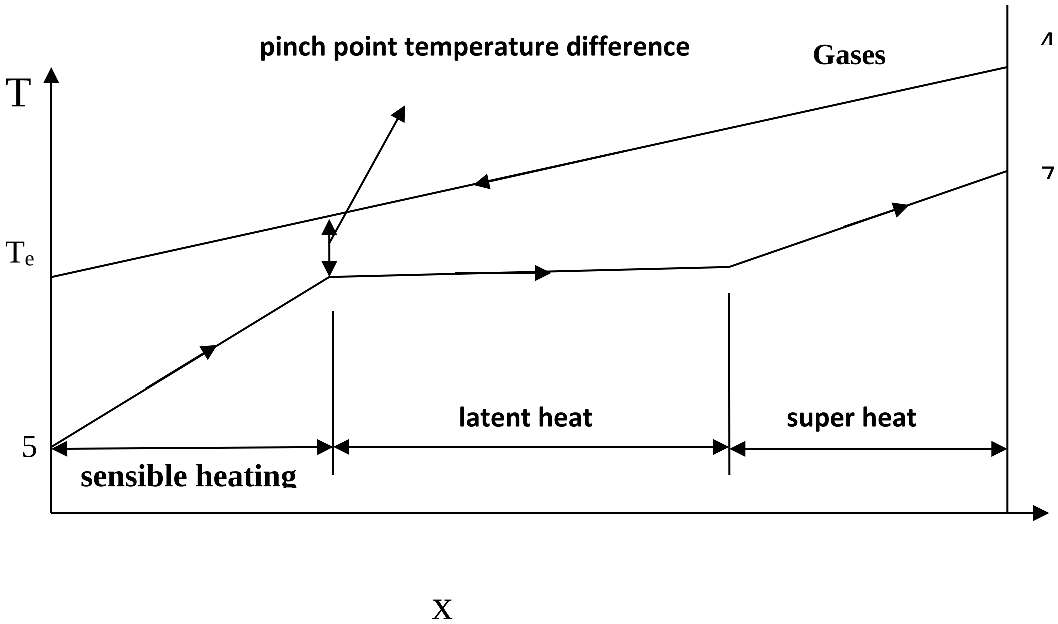


Fig 2.6 Heat transfer process between hot gases and the liquid undergoing phase change

CHAPTER 3

3 LITERATURE REVIEW

Thermodynamic analysis of power cycles have been the subject of intense study among researchers. Thermodynamic analysis involves a dual study of both energy and exergy based evaluation. Rankine cycle has been the driving horse of all steam based power plants over the years. Lot of modifications have been introduced into the basic Rankine cycle like regeneration and reheating to enhance its efficiency. Gas based power plants have also been injected in the arena of power generation since last two decades as new resources of natural gas have been explored. The idea of combined cycles surfaced in the midst of all these developments which was subjected to extensive studies as the literature review indicates. A combined cycle is a conglomeration of a Rankine cycle and a Brayton cycle. The authors feel that there is enough scope for further study in this area. A brief review of the studies done so far is as mentioned below.

Ali Bolattürk and Mehmet Kanoğlu [1] performed optimization studies of different types of power cycles. The optimization is performed to determine the optimum pressure ratios in gas-turbine cycles and the optimum boiler pressures in steam cycles that maximize the thermal efficiency of the cycle in energy method and the exergy efficiency in exergy method.

Tomasz Kowalczyk et.al [2] studied binary vapour cycle based on a model of a supercritical steam power plant. Energy analysis was done to conduct a preliminary optimization of the cycle. Exergy loss analysis was employed to perform a comparison of heat-transfer processes, which are essential for hierarchical cycles. The primary aim of this modification is to reduce the size of the power unit by decreasing the low-pressure steam turbine cylinder and the steam condenser.

Rice.I.G[3] presented an analysis of the simple and reheat gas turbine cycles and related these cycles to the combined gas turbine- Rankine cycle A unique arrangement of the super heater is discussed whereby part of the steam heat load is shifted to the reheat gas turbine to obtain a minimum heat recovery boiler stack temperature and a maximum cycle efficiency. This proposed power plant is projected to have a net cycle efficiency of 50 percent LHV when burning distillate fuel.

Harvey S. Leffa[4] conducted thermodynamic analysis of combined cycle power plants to exhibit how relatively straightforward thermodynamics, along with advances in the thermal properties of materials and clever design, can be used to dramatically increase the efficiency of electricity generation. Modern combined-cycle electric generating plants have changed the landscape of electric power generation. These combined-cycle power plants now reach thermal efficiencies in excess of 0.60. It is shown how the laws of thermodynamics make this possible.

Vundela Siva Reddy et.al [5] analyzes the information available in the open literature regarding energy and exergy analysis on high temperature power plant. A comprehensive literature review on thermal power plants, especially boiler in coal based thermal power plants and combustion chamber in gas-steam cogeneration was included. Finally, explaining the procedure of analysis of thermal power plant systems by exegetical approach was done.

S.C. Kaushik et.al.[6] used the second-law approach for the thermodynamic analysis of the reheat combined Brayton/Rankine power cycle. They reported that the exergy destruction in the combustion chamber represents over 50% of the total exergy destruction in the overall cycle. The combined\ cycle efficiency and its power output were maximized at an intermediate pressure-ratio, and increased sharply up to two reheat-stages and more slowly thereafter.

CoPadma Dhar Garg, et.al[7] state that the combined cycle cogeneration has the possibility to produce power and process heat more efficiently, leading to higher performance and reduced greenhouse gas emissions present work is to evaluate the performance of a combined cycle cogeneration configuration based on energy and exergy analyses approaches It is demonstrated that a combined cycle cogeneration unit, operates more efficiently and produces less carbon dioxide than two separate, power production and process heat systems The results pinpoint that more exergy losses occurred in the gas turbine combustion chamber reaching 35% of the total exergy losses, while the exergy losses in the other plant components are between 7% and 21% of the total exergy losses at 1400°C turbine inlet temperature and pressure ratio 10.

Based on the literature survey as shown the authors of this work have embarked on evaluation of combined Brayton-Rankine cycle system which adopts reheating.

CHAPTER 4

4 THERMODYNAMIC ANALYSIS

Statement of the Problem

To perform energy and exergy analysis of a Brayton-Rankine combined power cycle with reheating adopted and to further evaluate the effect of important parameters like pressure ratio of gas cycle, Gas turbine inlet temperature, steam inlet pressure ,reheat and condenser pressures etc., on its performance.

Assumptions

- The working fluid is assumed as a perfect gas and obeys all gas laws.
- The properties of the working fluid i.e. C_P, C_v, R& γ are constant and equal for both air and gases.
- The efficiencies of the individual components are assumed as η_{gt}=85%, η_{st}=80%, η_{cc}=90%, η_{hrsg}=90%, η_{comp}=85%, $\eta_{reheater}$=80%.
- The combustion process, reheating and heat transfer in HRSG are assumed as constant pressure process.
- The reheating process is assumed to take place from a constant high temperature source.
- The fuel used is octane with calorific value of 44,427kJ/kg and specific exergy of 47,346kJ/kg.
- The pinch point temperature difference is assumed as 15^0C.

4.1 BRAYTON CYCLE:

4.1.1 ENERGY EQUATIONS

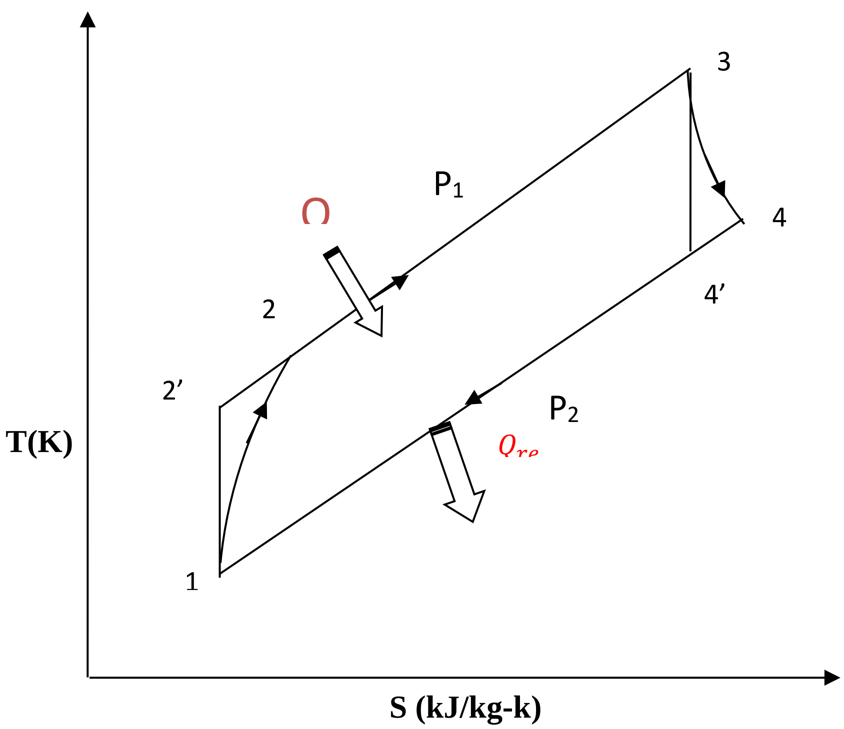


Fig 4.1 T-s plot of Brayton cycle

a) COMPRESSOR:

$$\frac{T_2{}'}{T_1} = (r_p)^{\frac{(\gamma-1)}{\gamma}} = \frac{P_2}{P_1}$$

Where r_p represents pressure ratio

$$h_2 = c_P(T_2)$$

$$W_C = h_2 - h_1$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} = \frac{h_2' - h_1}{h_2 - h_1}$$

b) COMBUSTION CHAMBER:

$$\dot{m}_a h_2 + m_{f1}(c.v) = m_g h_3$$

$$m_g = \dot{m}_a + m_{f1}$$

Where $\dot{m}_a$ reprents mass of air

m_{f1} reprents mass of fuel supplied in combustion chamber

c.v represents calorific value of fuel

m_g reprents mass of gas

c) TURBINE:

$$\frac{T_3}{T_4'} = \frac{P_3}{P_4} = (r_p)^{\frac{(\gamma-1)}{\gamma}}$$

$$W_{GT} = h_3 - h_4$$

Actual work = $h_3 - h_4$

$$\eta_{GT} = \frac{T_3 - T_4}{T_3 - T_4'} = \frac{h_3 - h_4}{h_3 - h_4'}$$

4.2 EXERGY ANALYSIS OF BRAYTON CYCLE

a) COMPRESSOR:

$$E.D_C = W_{act} - W_{max} = T_0(s_2 - s_1)$$

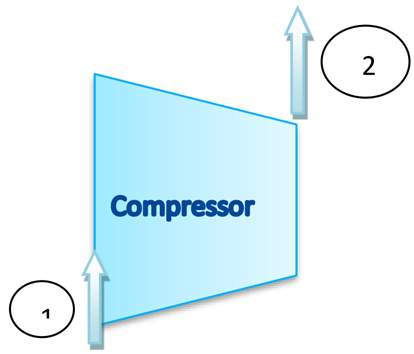


Fig 4.2 compressor

$$W_{act} = h_2 - h_1$$

$$P_{comp} = \dot{m}_a (h_2 - h_1)$$

$$W_{max} = e_2 - e_1$$

$$= (h_2 - h_1) - T_0(s_2 - s_1)$$

$$s_2 - s_1 = c_P \log\left(\frac{T_2}{T_1}\right) - R \log\left(\frac{P_2}{P_1}\right)$$

Where S_1 and h_1 represent the entropy and enthalpy at compressor inlet pressure

S_2 and h_2 represent the entropy and enthalpy at compressor outlet pressure

b) <u>COMBUSTION CHAMBER</u>:

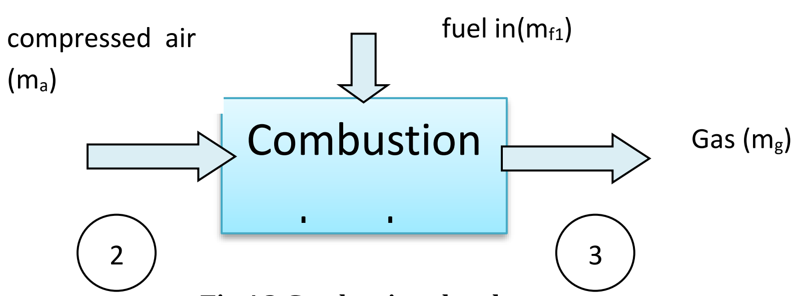


Fig 4.3 Combustion chamber

$Q_{input} = m_{f1}(c.v)$

$Enthalpy\ increase = m_g h_3 - \dot{m}_a h_2$

$$\eta_{c.c} = \frac{m_g h_3 - \dot{m}_a h_2}{m_{f1}(c.v)}$$

$m_g = m_{f1} + 1$

$$m_{f1} = \frac{h_3 - h_2}{\eta_{c.c}(c.v) - h_3} = \frac{c_p(T_3 - T_2)}{\eta_{cc}(c.v) - c_p T_3}$$

$E.D_{C.C} = E_{supplied} - E_{recovery}$

$E_{supplied} = m_{f1}(e_f)$

$$\begin{aligned} E_{recovery} &= E_3 - E_2 \\ &= m_g e_3 - m_a e_2 \\ &= (1 + m_{f1})h_3 - h_2 - T_0[(1 + m_{f1})s_3 - s_2] \end{aligned}$$

Where

$$s_3 = c_P \log\left(\frac{T_3}{T_0}\right) - R\ \log\left(\frac{P_2}{P_1}\right)$$

$$s_2 = c_P \log\left(\frac{T_2}{T_0}\right) - R\ \log\left(\frac{P_2}{P_1}\right)$$

c) GAS TURBINE :

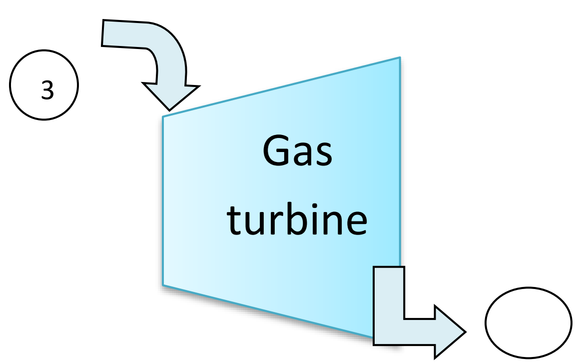


Fig 4.4 Gas turbine

$$W_{act} = h_3 - h_4 = \frac{c_p(T_3 - T_4')}{\eta_{GT}}$$

$$P_{gt} = \dot{m}_g\,(h_3 - h_4)$$

$$W_{max} = e_3 - e_4 = (h_3 - h_4) - T_0(s_3 - s_4)$$

$$s_3 - s_4 = c_P \log\left(\frac{T_3}{T_4}\right) - R\ \log\left(\frac{P_3}{P_4}\right)$$

where $P_1 = P_4,\ P_2 = P_3$

Exergy destruction in the gas turbine is given by,

$$E.D_{GT} = W_{max} - W_{act}$$

Where s_3 and h_3 represent entropy and enthalpy at gas turbine inlet pressure

d) <u>HEAT RECOVERY STEAM GENERATOR(HRSG):</u>

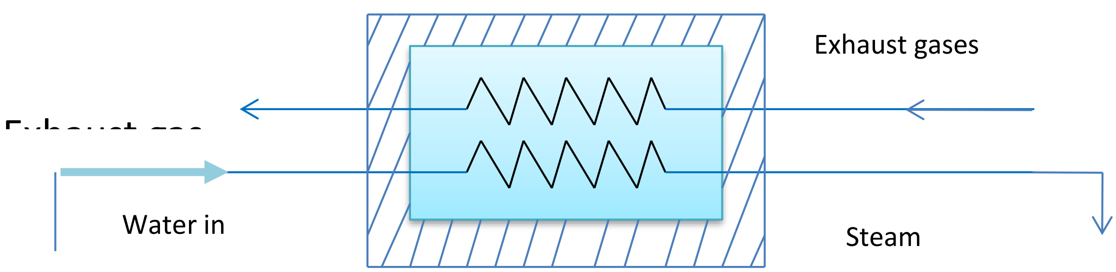


Fig 4.5 Heat Recovery Steam Generator

$$T_p - T_a = T_P - T_{sat}$$

Where T_p is the pinch point temperature

$$T_p = T_{sat} + 10\ to\ 15^0\ C$$

$$E_{supplied} = \dot{m}_g(e_4 - e_e)$$

$$= \dot{m}_g[(h_4 - h_e) - T_0(s_4 - s_e)]$$

$$= \dot{m}_g \left[c_P (T_4 - T_e) - c_p \log\left(\frac{T_4}{T_e}\right)\right]$$

$$\dot{m}_s = \frac{\eta_{hrsg}\,\dot{m}_g c_p (T_4 - T_e)}{h_7 - h_a}$$

Where m_s represent steam flow rate

$$E_{recovery} = \dot{m}_s (e_7 - e_5)$$

$$= (h_7 - h_5) - T_0 (s_7 - s_5)$$

$$s_5 - s_{tp} = c_P \log\left(\frac{T_5}{T_{tp}}\right)$$

where $\quad T_5 = c_p\, log\left(\frac{T_5}{T_{tp}}\right)$

Exergy destruction in heat recovery steam generator,

$$E.D_{hrsg} = E_{supplied} - E_{recovery}$$

$$\eta_{II} = \frac{E_{recovery}}{E_{supplied}}$$

4.3 EXERGY ANALYSIS OF RANKINE CYCLE WITH REHEATING

a) HIGH PRESSURE STEAM TURBINE :

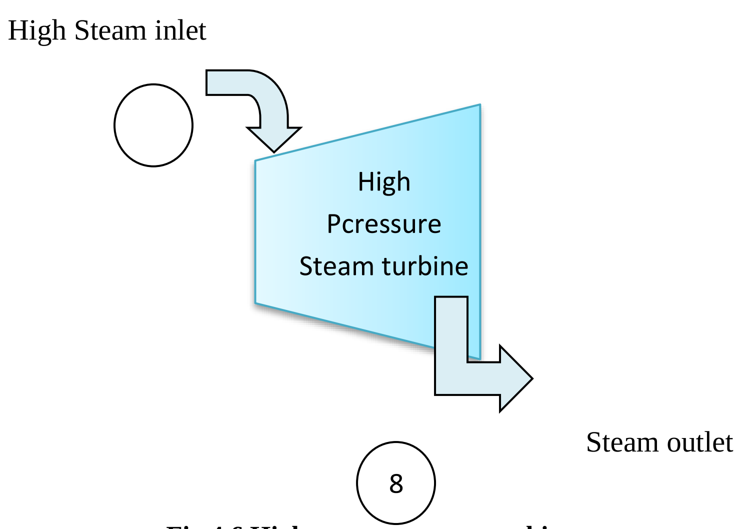


Fig 4.6 High pressure steam turbine

$$\eta_{ST1} = \frac{h_7 - h_8}{h_7 - h_8'}$$

$$h_8 = h_7 - \eta_{st}(h_7 - h_8')$$

$$W_{max} = \dot{m}_s(e_7 - e_8) = \dot{m}_s[(h_7 - h_8) - T_0(s_7 - s_8)]$$

$$W_{act} = \dot{m}_s(h_7 - h_8)$$

$$\eta_{II st1} = \frac{W_{act}}{W_{max}}$$

$$E.D_{st1} = W_{max} - W_{act}$$

Where $E.D_{st.1}$ represent Exergy destruction in high pressure steam turbine

b) REHEATER:

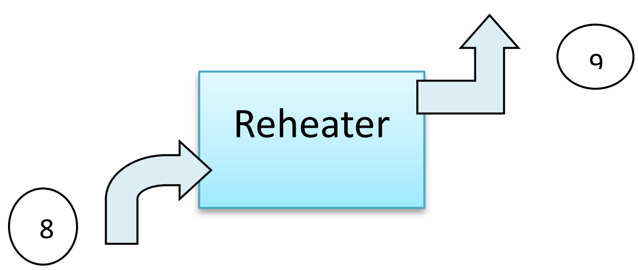


Fig 4.7 Reheater

$$Q_{gen} = \frac{\dot{m}_s(h_9 - h_8)}{\eta_{reheater}}$$

$$E_{recovery} = \dot{m}_s(e_9 - e_8) = \dot{m}_s[(h_9 - h_8) - T_0(s_9 - s_8)]$$

$$E_{supplied} = Q\left[1 - \frac{T_0}{T_{source}}\right]$$

$$E.D_{reheater} = E_{supplied} - E_{recovery}$$

$$\eta_{II\,reheater} = \frac{E_{recovery}}{E_{supplied}}$$

Where $E_{recovery}$, $E_{supplied}$ represents Exergy recovered and Exergy supplied in the reheater

T_0 represents source temperature

c) LOW PRESSURE STEAM TURBINE :

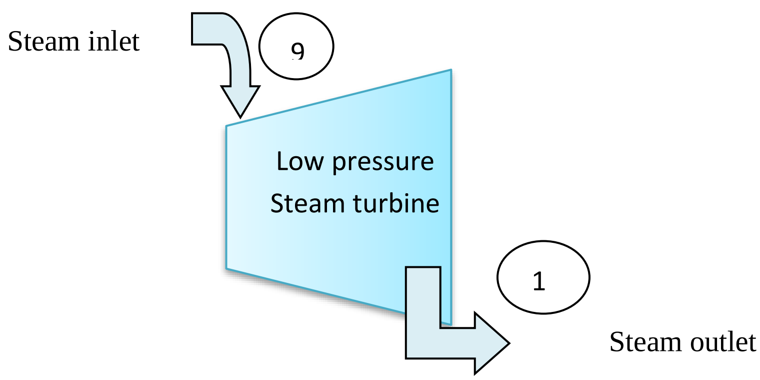


Fig 4.8 Low pressure steam turbine

$$\eta_{ST2} = \frac{h_9 - h_{10}}{h_9 - h_{10}'}$$

$$s_9 = s_{10}' = s_f + x_{10}' s_{fg}$$

$$x_{10}' = \frac{s_9 - s_f}{s_{fg}}$$

$$h_{10}' = h_f + x_{10}' h_{fg}$$

$$W_{act} = \dot{m}_s(h_9 - h_{10})$$

$$W_{max} = \dot{m}_s(e_9 - e_{10}) = \dot{m}_s[(h_9 - h_{10}) - T_0(s_9 - s_{10})]$$

$$E.D_{st2} = W_{max} - W_{act}$$

$$\eta_{IIst2} = \frac{W_{act}}{W_{max}}$$

Where S_9 and h_9 represents entropy and enthalpy at low pressure steam inlet pressure

S_{10} and h_{10} represents entropy and enthalpy at low pressure steam outlet pressure

d) STEAM CONDENSER:

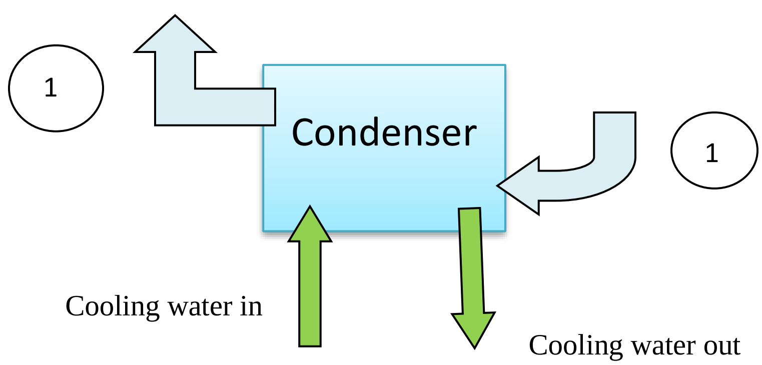


Fig 4.9 Steam condenser

$$E_{recovey} = 0$$

$$E_{supplied} = \dot{m}_s(\,e_{10} - e_{11}) = \dot{m}_s[(h_{10} - h_{11}) - T_0(s_{10} - s_{11})]$$

$$s_{11} = s_f \; at \; P_5$$

$$E.D_{conenser} = E_{supplied} - E_{recovery}$$

e) FEED PUMP:

$$\eta_{feed\ pump} = \frac{h_5' - h_{11}}{h_5 - h_{11}}$$

$$h_5' - h_{11} = V_{avg} \; dp$$

$$V_{avg} = \frac{v_f \; at \; P_3 + v_f \; at \; P_5}{2}$$

$$h_5 = h_{11} + \frac{h_5' - h_{11}}{\eta_{feed\ pump}}$$

$$W_{act} = \dot{m}_s(h_5 - h_{11})$$

$$W_{min} = \dot{m}_s(\,e_5 - e_{11}) = \dot{m}_s[(h_5 - h_{11}) - T_0(s_5 - s_{11})]$$

$$\eta_{II\,feed\ pump} = \frac{W_{min}}{W_{act}}$$

f) EXHAUST GASES:

Exergy carried away by the exhaust gases

$$E.D_{waste\ gases} = \dot{m}_g \; c_p \left[\left(T_g - T_0\right) - T_0 \, log\left(\frac{T_g}{T_0}\right)\right]$$

4.4 OVERALL EXERGY DESTRUCTION:

$$E.D_{TOTAL} = E.D_{comp} + E.D_{c.c} + E.D_{gt} + E.D_{hrsg} + E.D_{st-1} + E.D_{rehater} + E.D_{st-2} + E.D_{cond} + E.D_{f.p} + E.D_{waste\ gases}$$

$$E_{total\ supplied} = \dot{m}_f(e.f) + Q\left(1 - \frac{T_0}{T_{source}}\right)$$

$$E_{recovery} = E_{total\ supplied} - E.D_{total}$$

$$Q_{toatl\ heat\ supplied} = \dot{m}_{f1}(\ c.v) + Q_{gen}$$

$$Overall\ power_{net} = P_{net-1} + P_{net-2}$$

$$where\ P_{net-1} = \ W_{gt} - W_{c}$$

$$P_{net-1} = \ W_{st-1} + W_{st-2} - W_{f.p}$$

4.5 EFFICIENCIES:

$$\eta_{exergy} = \frac{Overall\ Power_{net}}{E_{total\ supplied}}$$

$$\eta_{energy} = \frac{Overall\ Power_{net}}{Q_{total\ heat\ supplied}}$$

Power ratio: It is defined as the ratio power output of gas cycle power output of steam cycle

$$Power\ ratio = \frac{P_{net-1}}{P_{net-2}}$$

CHAPTER 5

5 RESULTS AND DISCUSSION

The following operating parameters are chosen for analysing the cycle.

5.1 OPERATING PARAMETERS

A) Brayton cycle

Air inlet temperature to compressor T_1: 303 K

Mass flow rate of air (m_a) : 1 kg/s

Compressor efficiency : 70 - 90 %

Pressure ratio : 1-10

Maximum temperature in Brayton cycle : 1273K - 1473K

Compressor efficiency: 70 to 80%

Gas turbine efficiency: 65 to 82%

Heat recovery steam generator efficiency:75 to 95%

B) **Rankine cycle**

Highest steam pressure:50 to 90 bar

Reheat steam pressure:10 to 40 bar

Condenser steam pressure:0.4 to 0.8 bar

5.2 PARAMETRIC ANALYSIS

5.2.1 EFFECT OF MAXIMUM TEMPERATURE OF GAS CYCLE

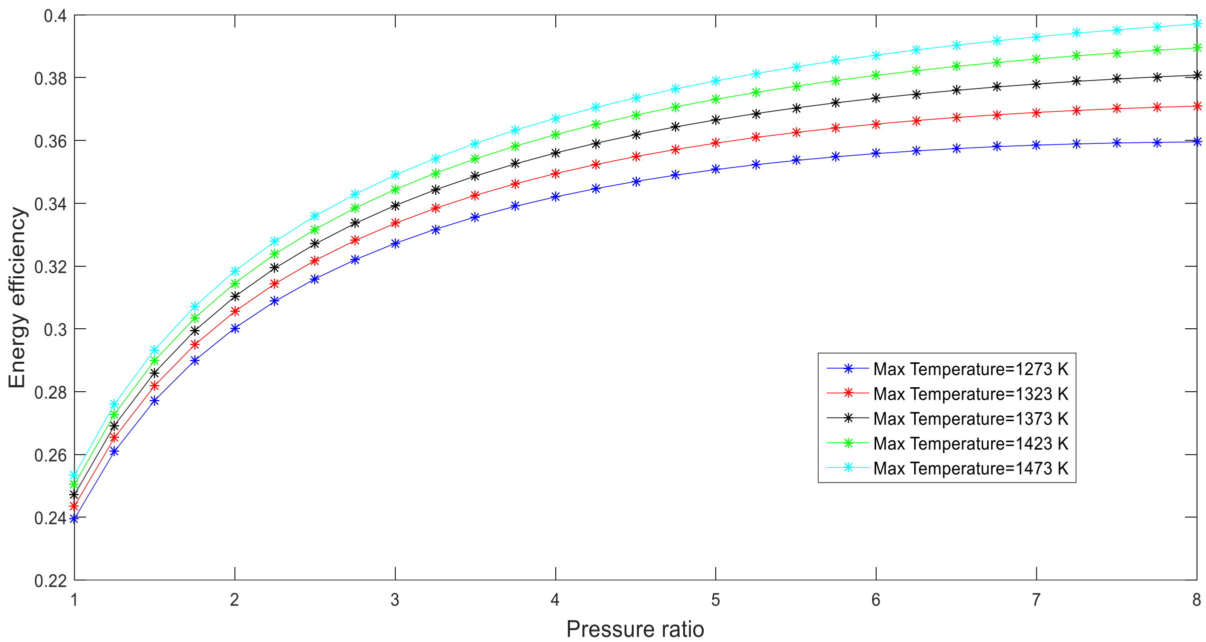


Fig 5.1 .Energy efficiency Vs. Pressure ratio with variation of maximum temperature

Parametric analysis plays a vital role in proper selection of parameters where the performance of the power plant is at its peak. The maximum temperature and pressure ratio of the gas power cycle are found to be the vital clogs in deciding the performance. It has been observed that there is limitation on the maximum temperature on the lower side for a given maximum pressure ratio. The pressure ratio considered for analysis are from 1 to 8 and the maximum temperatures are varies from 1000^0C to 1200^0C. From the plots 5.1 and 5.2, it can be observed that the increase in pressure ratio produces an increase in both energy and exergy efficiencies at a given peak temperature. The maximum temperature when enhanced creates a positive impact in the cycle. The maximum energy efficiency occurs at a pressure ratio of 8 and maximum temperature of 1200^0C, which are the maximum values of these parameters as considered for the cycle and the corresponding exergy efficiency is 38.14%. It is further observed that for a pressure ratio of unity, there is no power output from the gas cycle. Hence the efficiency from the plots represent only the efficiencies of steam cycle.

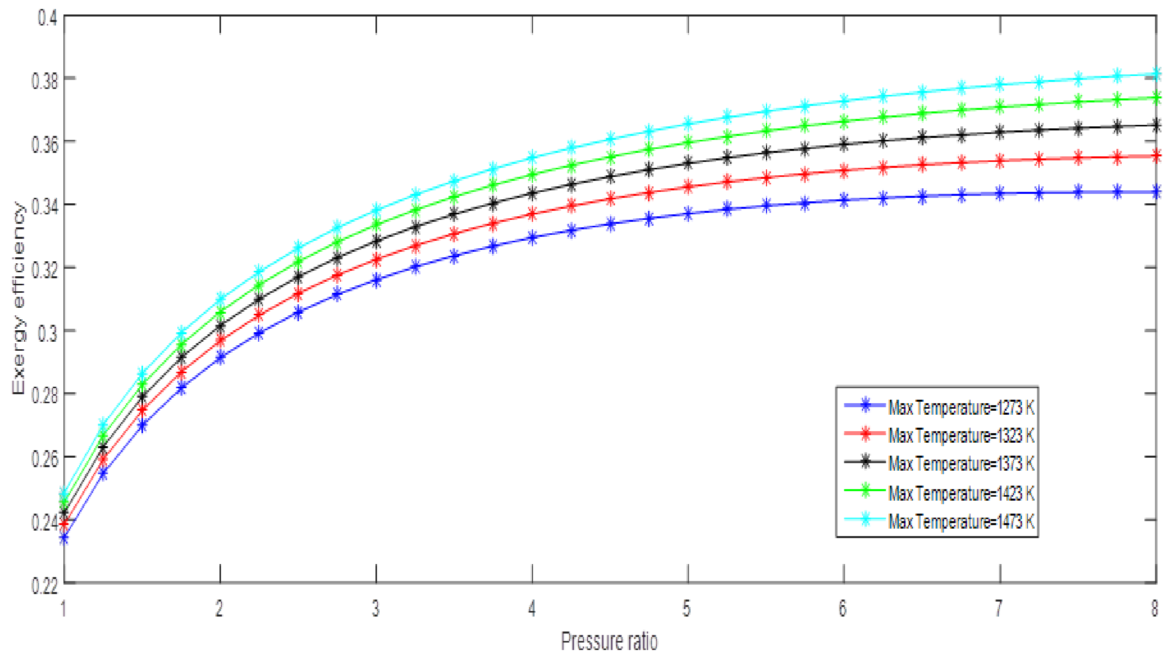


Fig 5.2 Exergy efficiency Vs. Pressure ratio with variation of maximum temperature

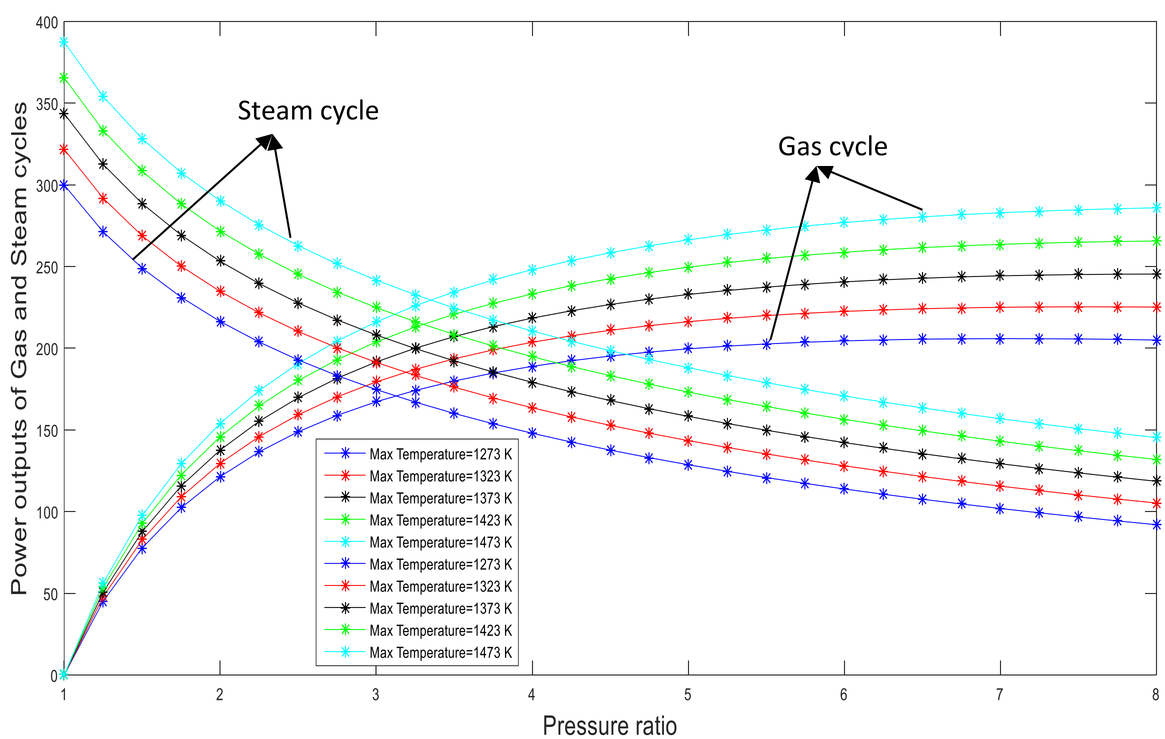


Fig 5.3 Power outputs of Gas and steam cycles Vs. Pressure ratio with variation of maximum temperature

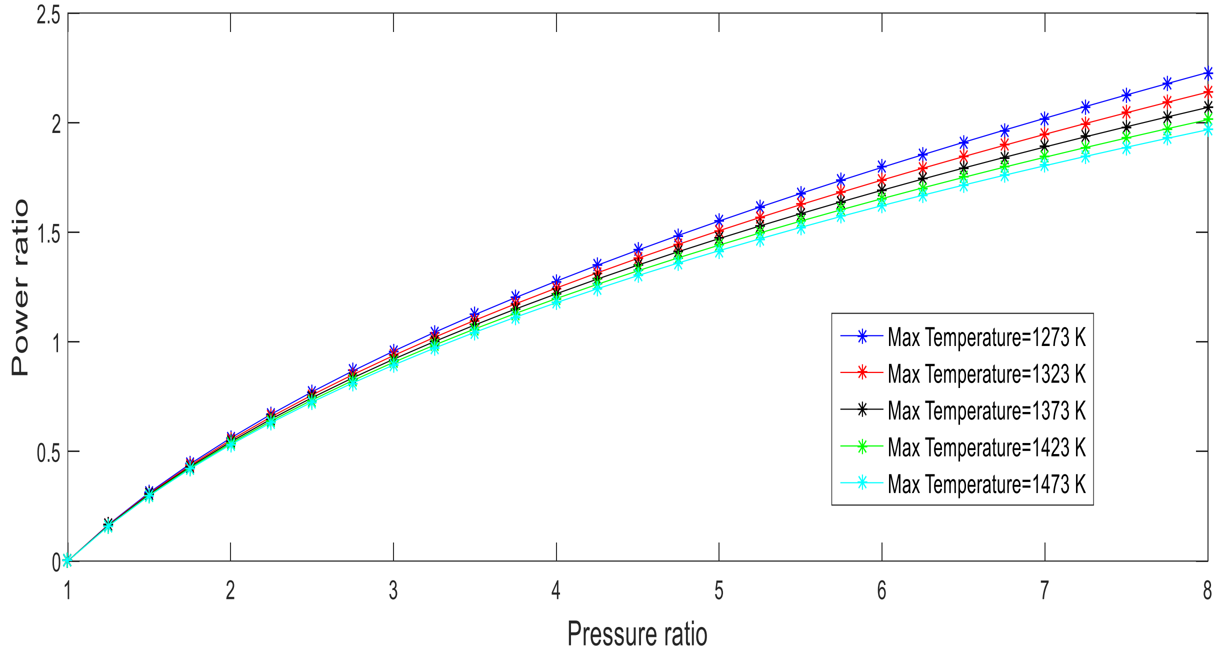


Fig 5.4 Power ratio Vs. Pressure ratio with variation of maximum temperature

In fig.5.3, the variation of power output from both gas and steam cycles have been plotted with respect to pressure ratio with maximum temperature taken as parameter. The ratio of the power outputs are plotted in fig5.4. It can be observed that the power output of gas turbine increases with pressure ratio as customary. The power output being higher at higher temperature for both gas and steam cycles. The steam cycle produces higher outputs at lower pressure ratio's and shows a declining trend with increase in pressure ratio. Both the cycles produce equal outputs in the pressure ratio range of 3 to 3.5.

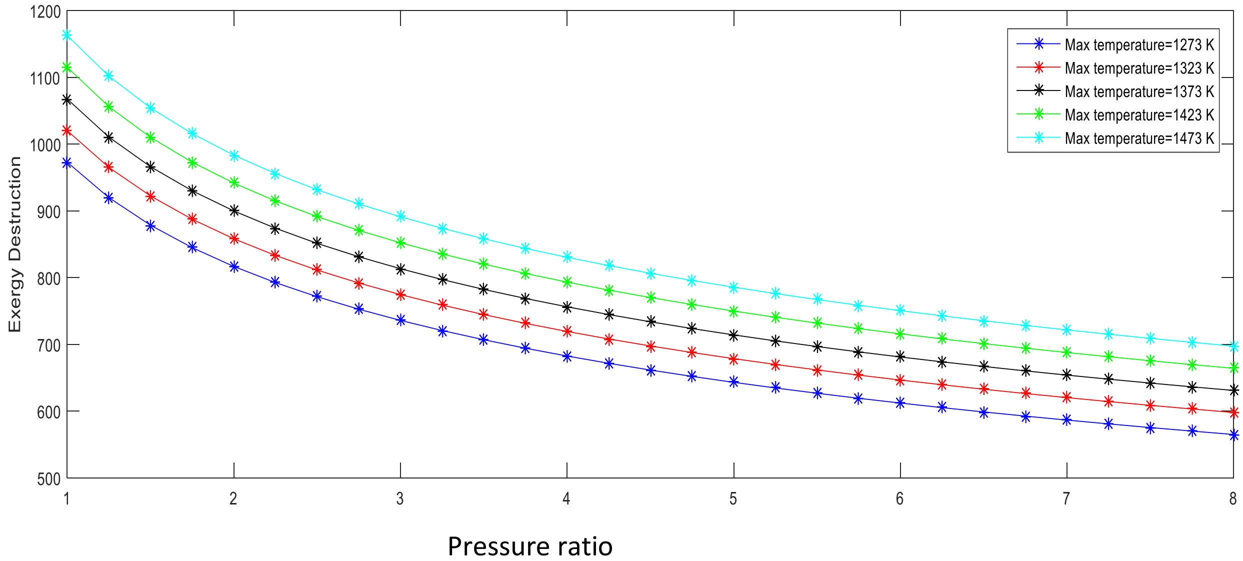


Fig 5.5 Exergy destruction Vs.Pressure ratio with variation of maximum temperature

The exergy destruction of the cycle is a function of the maximum temperature and pressure ratio. The higher the maximum temperature of the cycle, the higher is the exergy destruction in the combustion chamber, which forms the major part of the total exergy destruction. With increase in pressure ratio the exergy destruction in combustion chamber decreases. Hence the total exergy destruction shows a declining trend at higher pressure ratios.

To study the effect of energy efficiencies of the individual equipments on the performances of cycle, the compressor & turbine efficiencies were considered for parametric analysis.

5.2.2 EFFECT OF GAS TURBINE EFFICIENCY:

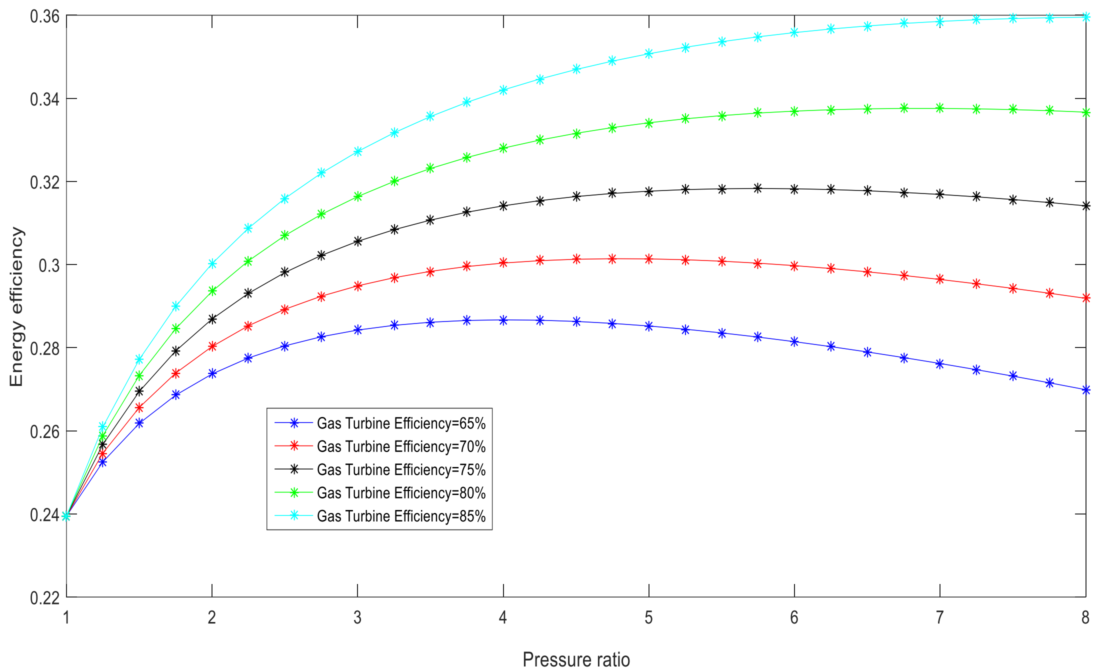


Fig 5.6 Energy efficiency Vs. Pressure ratio with variation of gas turbine efficiency

Interesting facts have come to the fore when gas turbine efficiency is taken up for parametric analysis. There seems to exist an optimum pressure ratio for every gas turbine efficiency considered. This optimum pressure ratio is found to be differing for the maximum energy and exergy efficiencies. The optimum pressure ratio for different turbine efficiencies is as given in the table

Table 5.1 The optimum pressure ratio for different turbine efficiencies is as given in the table

S.No.	Gas turbine efficiency	Optimum pressure ratio and corresponding maximum efficiency			
		Optimum pressure ratio	Energy efficiency	Optimum pressure ratio	Exergy efficiency
1	65	4.25	28.67	4.25	27.78
2	70	5	30.14	4.75	29.13
3	75	6	31.83	5.75	30.67
4	80	7	33.76	6.75	32.42

5	85	8	35.95	8	34.4

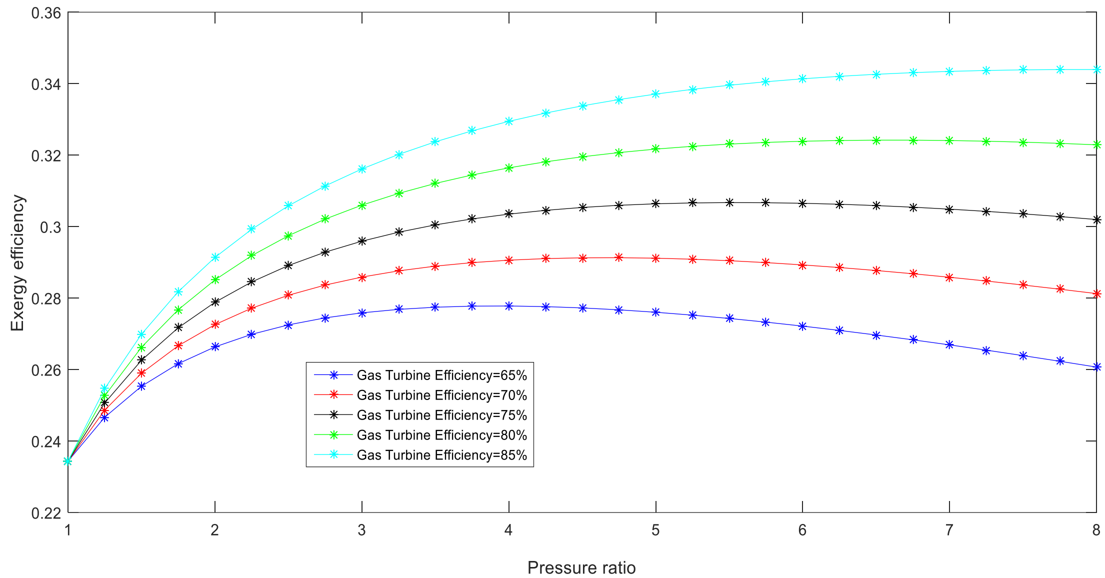


Fig 5.7 Exergy efficiency Vs. Pressure ratio with variation of gas turbine efficiency

The optimum pressure ratio for the maximum energy and exergy efficiencies are almost equal. It is also observed that the optimum pressure increments by almost 1 for every 5% increase in gas turbine efficiency as shown in the plots5.6 and 5.7.

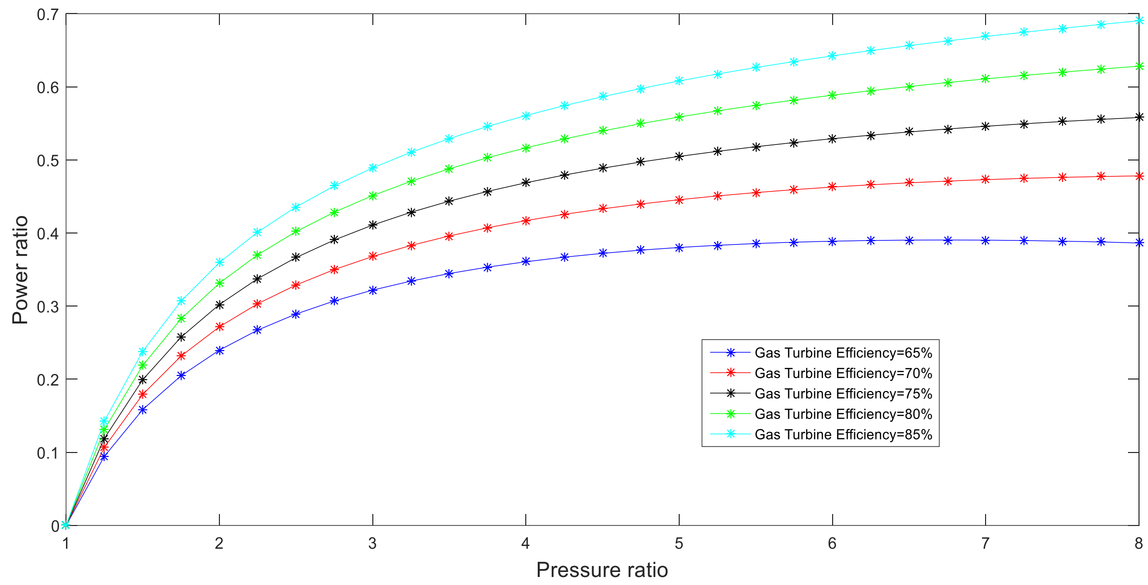


Fig 5.8 Power ratio Vs. Pressure ratio with variation of gas turbine efficiency

The power ratio is plotted with respect to pressure ratio for varying gas turbine efficiency in fig 5.8. The graph suggests that at lower gas turbine efficiencies, the steam cycle output forms the major part of the total power output. However with increase in pressure ratio the gas turbine output enhances and forms almost 64% of the steam plant output. This value increases with increase in gas turbine efficiency and its output almost reaches 91% of steam plant power output. It is further observed that with increase in pressure ratio the steam turbine output keeps pace with steam plant output and even crosses it at pressure ratios in excess of 5. With higher turbine performance, the output of gas cycle is more than twice the output of steam cycle.

5.2.3 EFFECT OF COMPRESSOR EFFICIENCY:

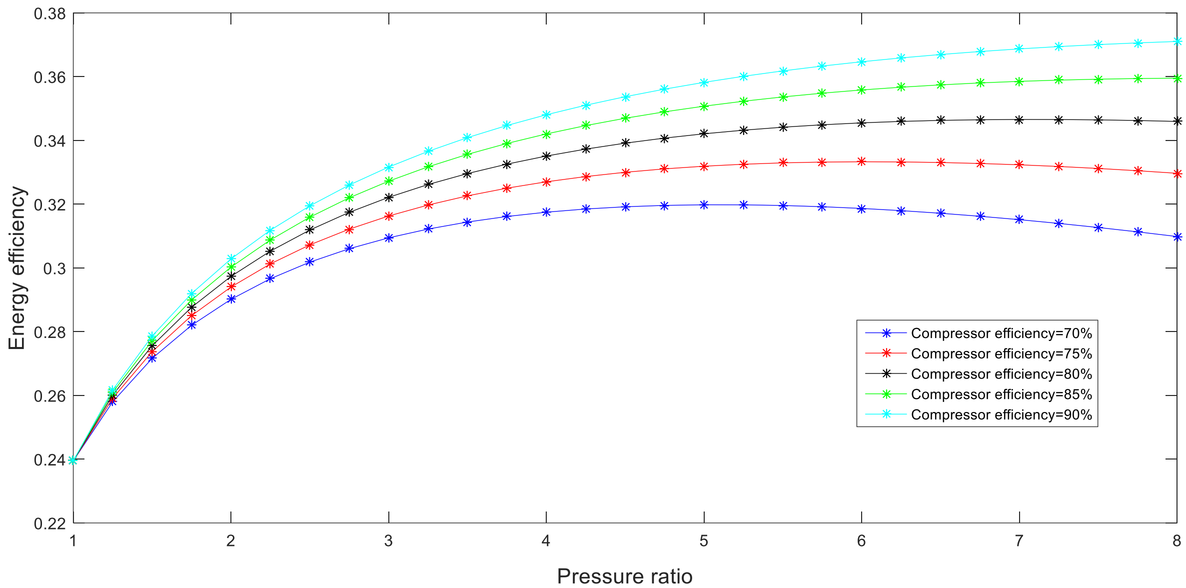


Fig 5.9 Energy Vs. Pressure ratio with variation of compressor efficiency

The variation of compressor efficiency brings out the optimum pressure ratio for maximum 1st & 2nd law efficiencies as indicated in fig 5.9. For a compressor efficiency of 70%, the optimum pressure ratio is 5 and it increases to 6 for a compressor efficiency of 75%. The corresponding maximum energy efficiencies are 31.98% & 33.33%. The optimum pressure ratio increases in steps of 1 for every 5% increase in compressor efficiency. Similar trends were also observed with regards to exergy efficiency as observed from fig 5.10.

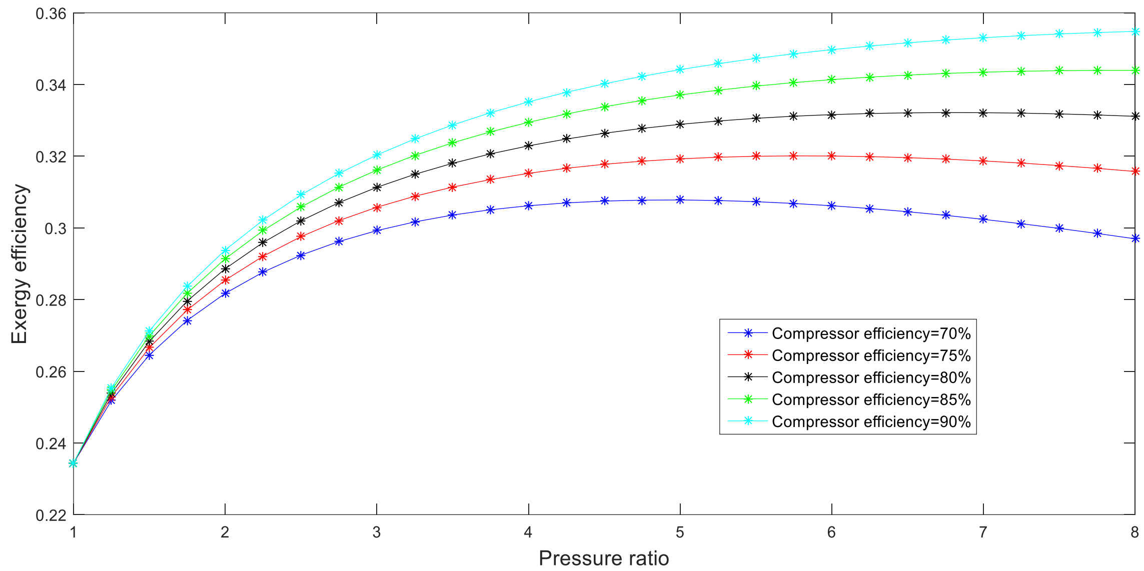


Fig 5.10 Exergy Vs. Pressure ratio with variation of compressor efficiency

The rate of steam generation is directly proportional to flow rate of gases which can be inferred from the energy balance of HSRG. With higher compressor efficiency, the fuel consumption rate increases, which will enhance the flow rate of gases. The increasing flow rate of gases will produce a corresponding increase in rate of steam flow. Hence ratio of flow rates of gases & steam remains at a steady value, irrespective of the compressor efficiency. With increasing pressure ratio, the fuel rate & gas flow rate drops. It is observed that the decreasing gas flow rate, decreases the steam flow rate. The decreasing steam flow rate is found to be slightly higher than decrease in gas flow rate. Hence the ratio of flow rates of gases & steam flow rate increases.

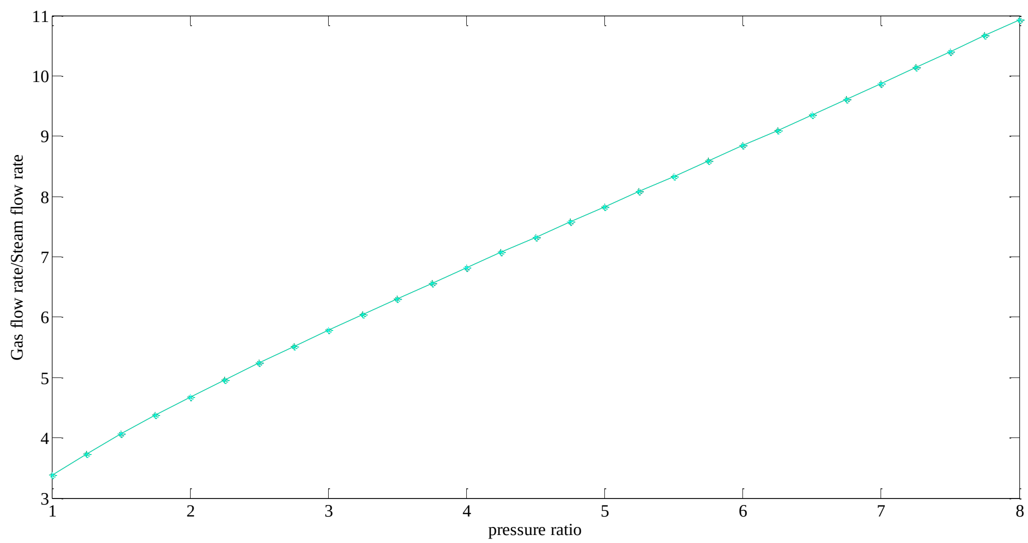


Fig 5.11 Pressure ratio Vs. Gas flow rate/steam flow rate with variation of compressor efficiency

5.2.4 EFFECT OF HEAT RECOVERY STEAM GENERATOR EFFICIENCY:

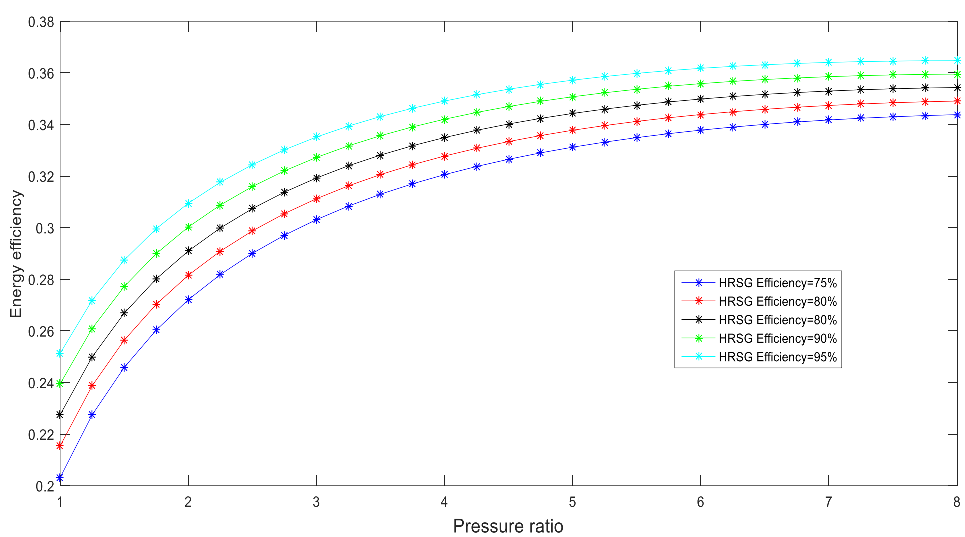


Fig 5.12 Energy efficiency Vs. Pressure ratio with variation of HRSG efficiency

The 1st law efficiency of heat recovery steam generator plays a pivotal role in heat recovery process. The higher its efficiency, the better will be the overall efficiency of the cycle as indicated in fig 5.12. Its effect on the cycle efficiencies is considered under varying pressure ratio. It is observed that with increasing pressure ratio both the energy and exergy efficiencies increase and reach asymptotic values. It is observed that the rate of increasing efficiency becomes less at higher pressure ratio values from fig 5.12&5.13.

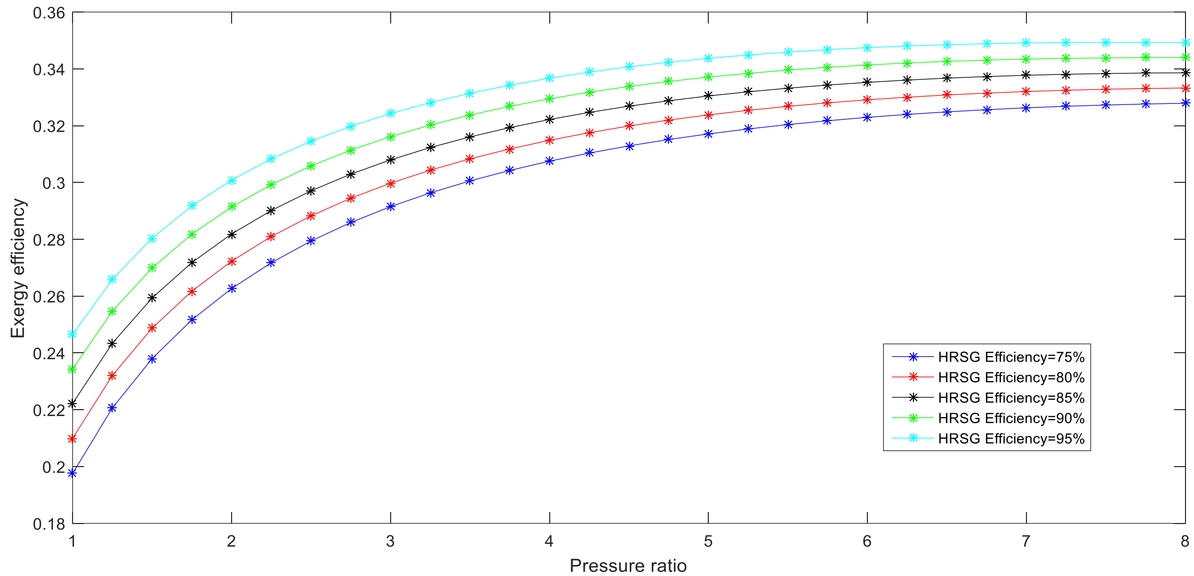


Fig 5.13 Exergy efficiency Vs. Pressure ratio with variation of HRSG efficiency

5.2.5 EFFECT OF HIGHEST STEAM PRESSURE:

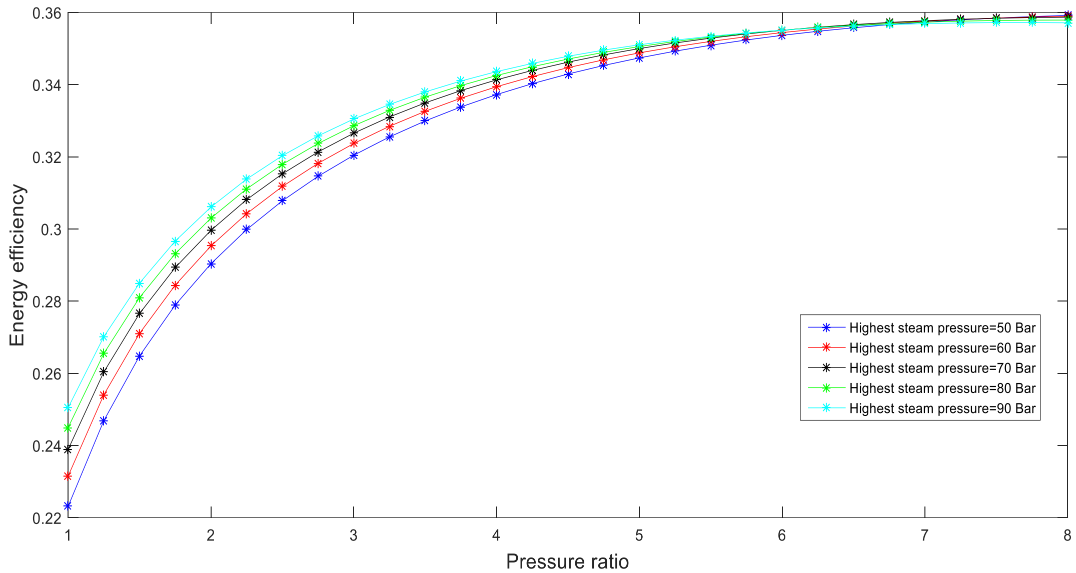


Fig 5.14 Energy efficiency Vs. Pressure ratio with variation of Highest steam inlet pressure

Interesting facts have been revealed while studying the effect of inlet pressure to turbine-1. Through it appears that increasing steam pressure will enhance the efficiency at a first glance which may be true for a normal Rankine cycle, it is necessary to study the effect of steam pressure in combination with Pressure ratio of gas turbine cycle. At lower pressure ratios of gas cycle adopting highest steam pressures seems to be beneficial with reference to 1st law efficiency as indicated in fig5.14.In the Pressure ratio range between 6 to 7 and the energy efficiency of the cycle seems to be independent of the highest steam pressure. The efficiency of cycle in that range is hovering around 35.4 to 35.77% .

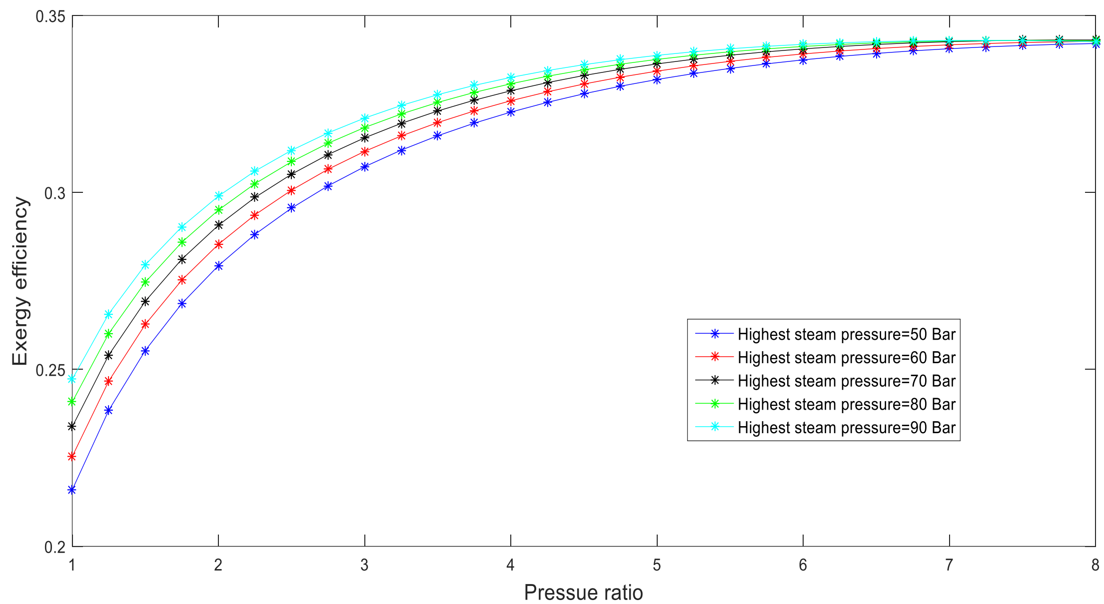


Fig 5.15 Exergy efficiency Vs. Pressure ratio with variation of Highest steam inlet pressure

A similar behaviour is also observed with regards to exergy efficiency. With increasing pressure ratio, the exergy efficiency of cycle seems to be independent of the maximum steam pressure. At lower pressure ratios of gas cycle, highest steam pressure seem to be advantageous.

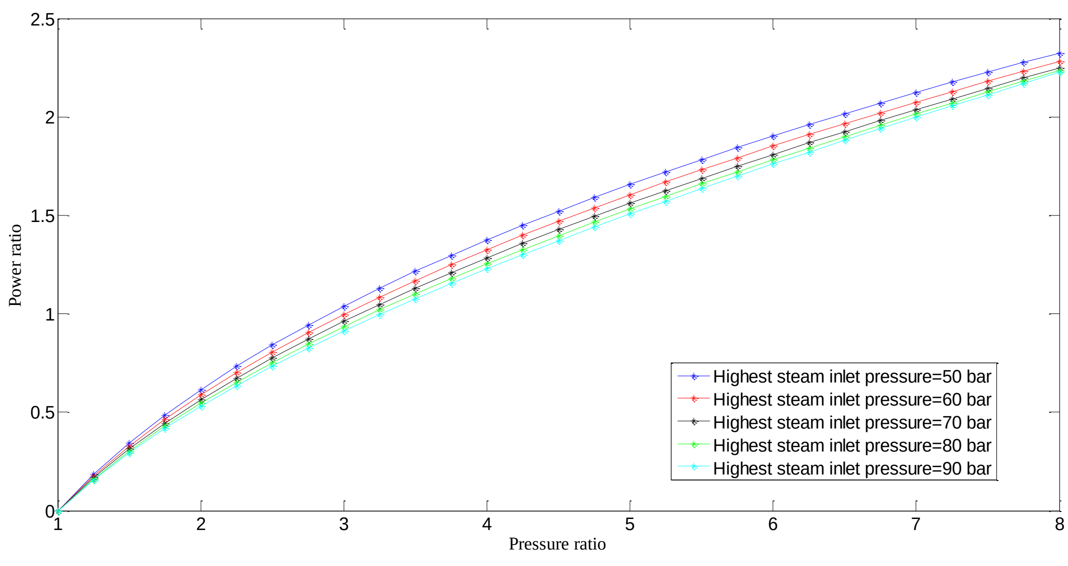


Fig 5.16 Power ratio Vs. Pressure ratio with variation of Highest steam inlet pressure

The gas turbine power output is observed to increase with increase in pressure ratio and reaches an optimum value of 205.8655 KW at a pressure ratio of 7. The power output of Rankine cycle decreases with increase in pressure ratio of the topping cycle. The power ratio therefore increases with increase in pressure ratio, but the effect of steam pressure diminishes at higher Brayton cycle pressure ratios.

5.2.6 EFFECT OF REHEAT STEAM PRESSURE:

The effect of reheat steam pressure has been taken as parameter to evaluate its effect on both energy and exergy efficiencies. The optimum reheat steam pressure for maximum energy efficiency is found to be about 35 bar, i.e. 50% of maximum steam pressure.

The maximum energy efficiency is 36.01% which occurs at pressure ratio of 8. It was interestingly observed that the maximum exergy efficiency requires further expansion of steam close to 15bar. The maximum exergy efficiency reaches a constant value 34.44% at a pressure ratio of 7.75 as indicated in figs.5.17&5.18.

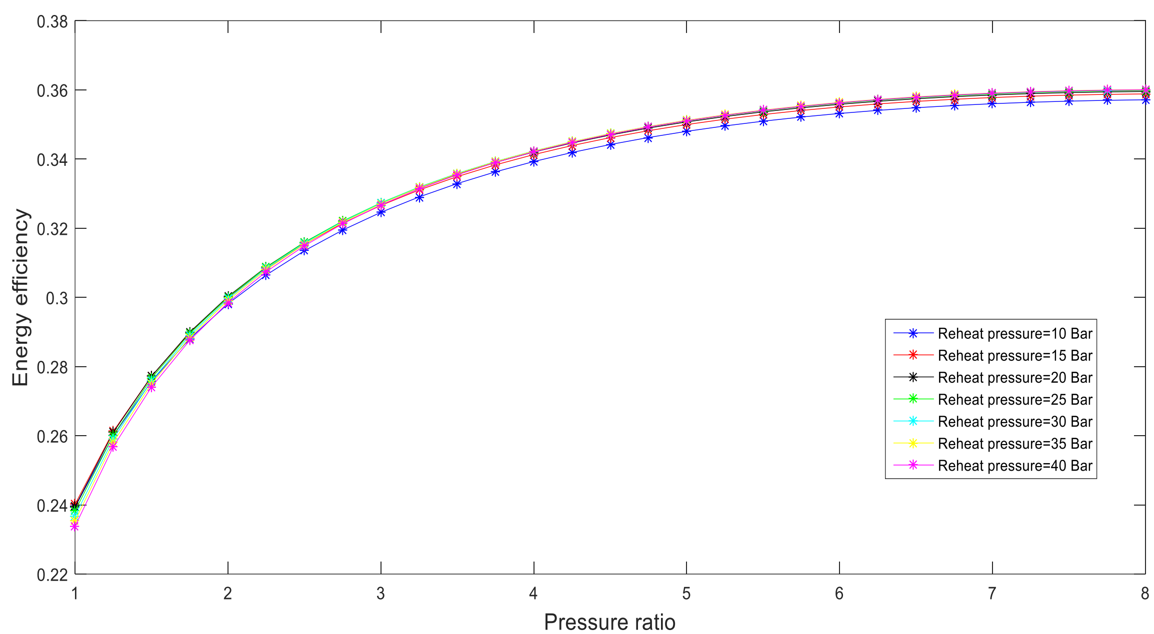


Fig 5.17 Energy efficiency Vs. Pressure ratio with variation of Reheat steam pressure

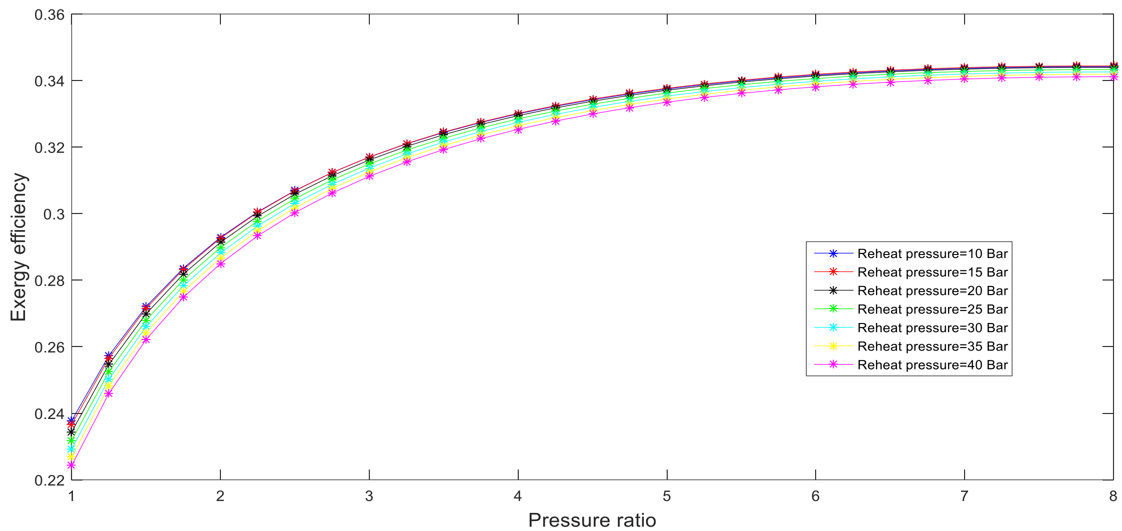


Fig 5.18 Exergy efficiency Vs. Pressure ratio with variation of Reheat steam pressure

The effect of reheat steam pressure on the net power output of both gas cycle and steam cycles is also studied as shown in fig5.19 &5.20. The gas turbine cycle as is known uninfluenced by reheat pressure, as it is an operating parameter of the bottoming cycle. However it is observed that the optimum pressure ratio of gas turbine cycle for the peak power output is 7.

The power output of the steam cycle is strongly dependent on the pressure ratio of the gas cycle and also the reheat steam pressure. It is observed that with increasing reheat pressure and also pressure ratio, the power out of the steam cycle decreases. This variation can be attributed to fact that the gas flow rate and steam flow rate decrease with increasing pressure ratio.

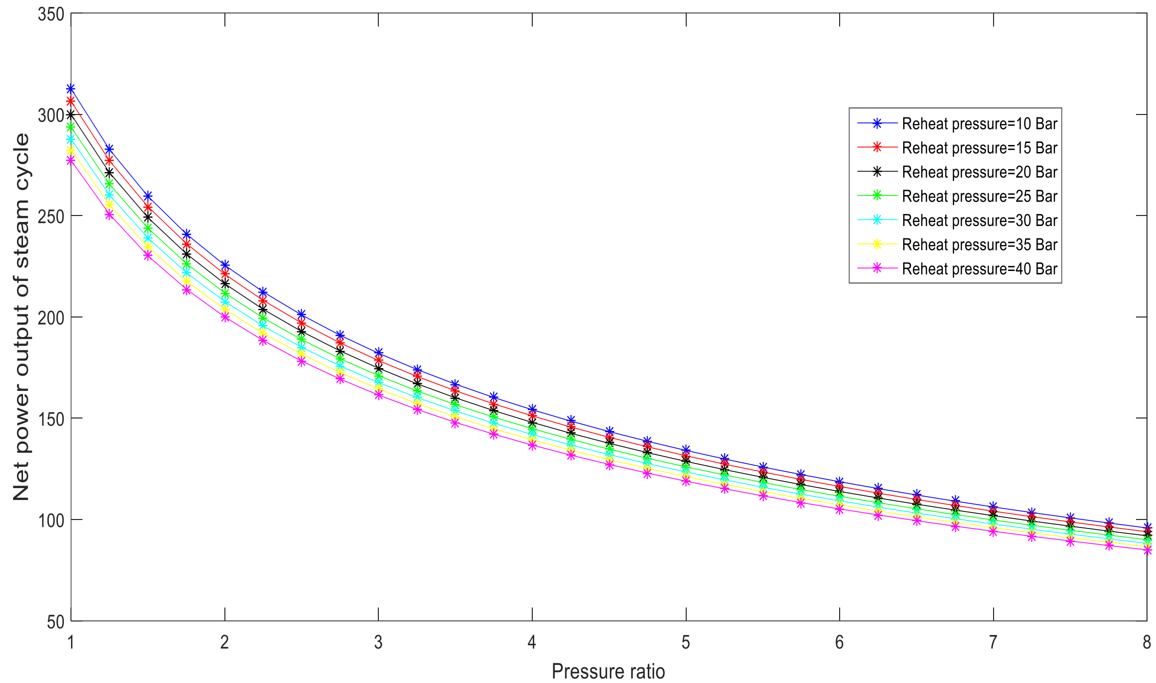


Fig 5.19 Net power output of steam cycle Vs. Pressure ratio with variation of Reheat steam pressure

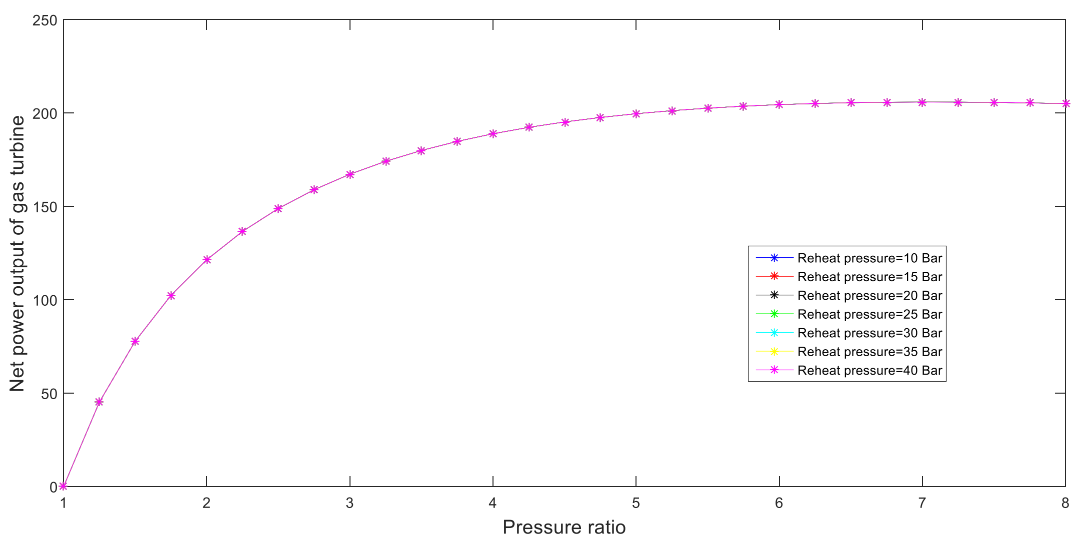


Fig 5.20 Net power output of gas cycle Vs. Pressure ratio with variation of Reheat steam pressure

5.2.7 EFFECT OF CONDENSER PRESSURE:

The condenser pressure was varied from 0.4bar to 0.8 bar (vacuum pressures). It is obvious that lower condenser pressures will promote higher efficiency values. But the interesting fact is that the enhancement in performance is diminishing at higher pressure ratios of gas turbine cycle. The efficiency of the cycle in terms of both energy and exergy are reaching maximum asymptotic values at a pressure ratio of around 8.The maximum energy efficiency is 36.27 % and maximum exergy efficiency is 34.7 % as shown in figs.5.21&5.22.

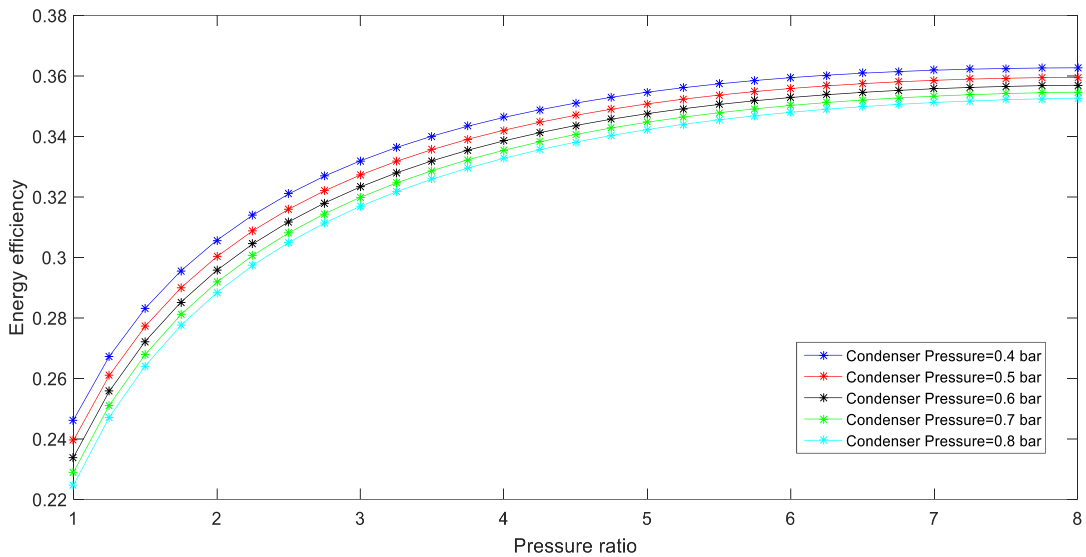


Fig 5.21 Energy efficiency Vs. Pressure ratio with variation of Condenser steam pressure

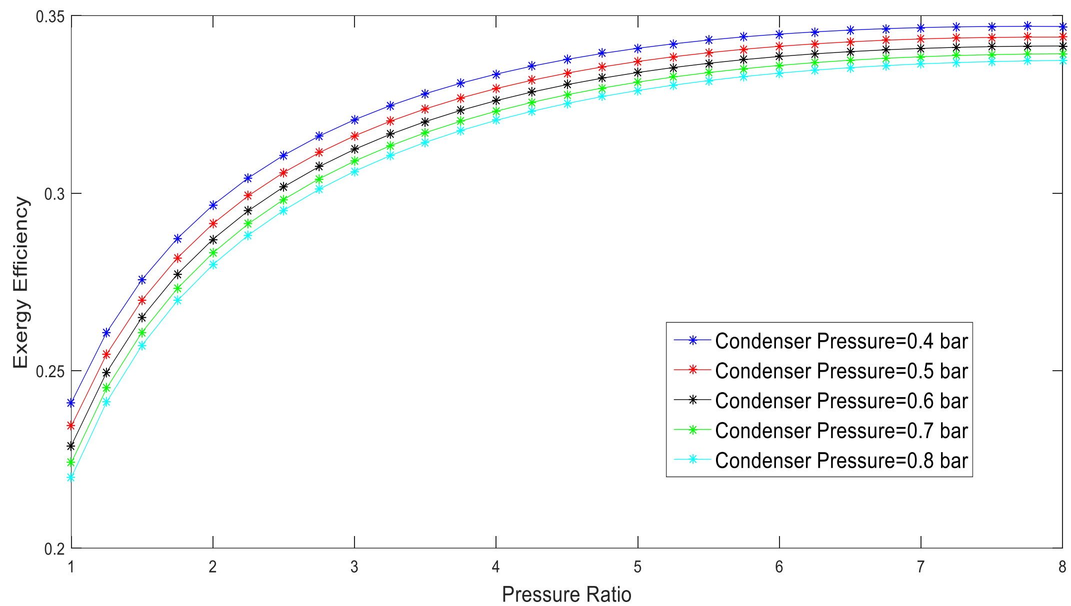


Fig 5.22 Exergy efficiency Vs. Pressure ratio with variation of Condenser steam pressure

The effect of condenser pressure along with pressure ratio of topping cycle on the power ratio is also analysed. The power ratio is found to increase with increase in pressure ratio at a given condenser pressure. It is further observed that with increasing condenser pressure, the power ratio also increases at a given pressure ratio. The enhancement of power ratio with increase in pressure ratio is found to be independent of condenser pressure. The highest power ratio is 2.37 which occurs at a condenser pressure of 0.8 bar and the minimum power ratio occurs at the lowest condenser pressure and pressure ratio as observed from fig.5.23.

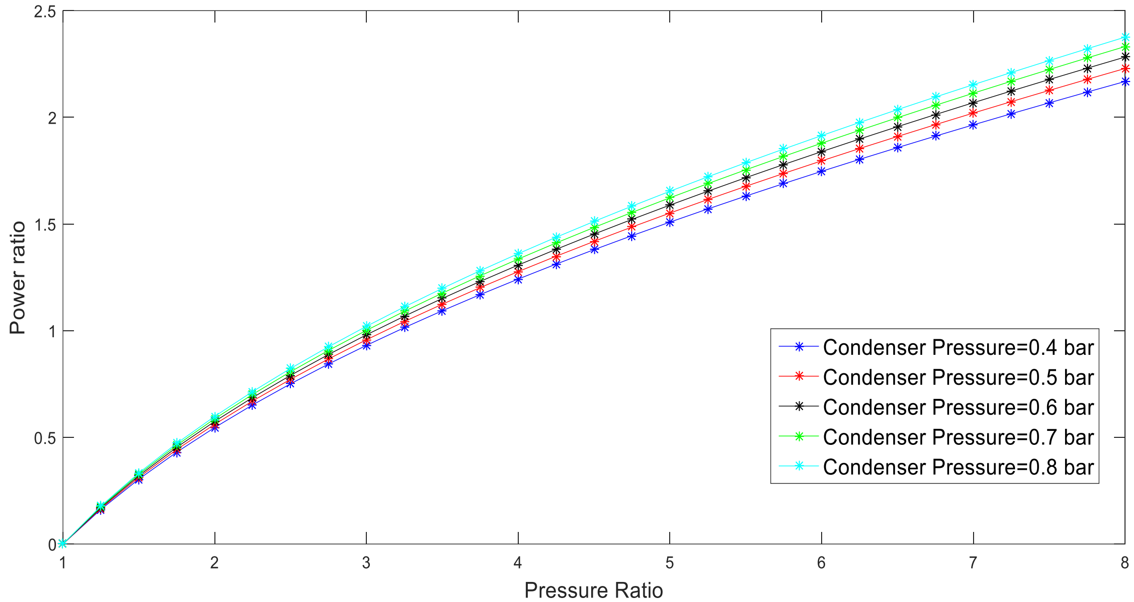


Fig 5.23 Power ratio Vs. Pressure ratio with variation of Condenser steam pressure

5.3 EXERGY DESTRUCTION IN INDIVIDUAL COMPONENTS

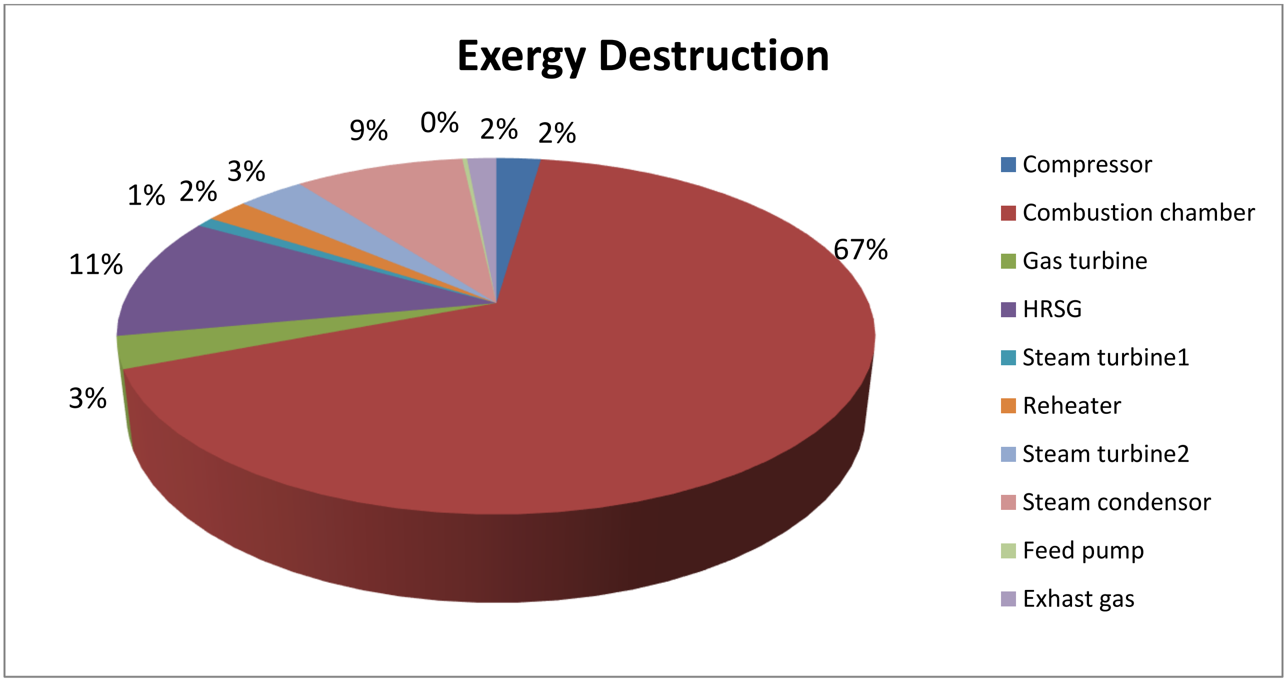


Fig 5.24 Pie diagram exhibiting percentage of exergy destruction in individual components for a particular operating condition(Tmax=1373K,rp=8,Pmax=70bar,Tsteam=773K, Preheat=20 bar.Pcond=0.5 bar).

The exergy destruction in the individual equipments gives an idea of existence of irreversibilities in this cycle. The major share of exergy destruction(67%) occures in the combustion chamber. The remaining 33% of exergy destruction in the cycle is distributed among all the components. The next major share of exergy destruction occurs in HRSG is 11% due to irreversibile heat transfer between the water and gas. Steam condenser contributes to 9% of total exergy destruction. There is no exergy recovery in the steam condenser. It is also observed that the exergy destruction in the low pressure turbine (25.4W) is almost four times the exergy destruction of high pressure steam turbine (6.76W). The total exergy destruction in the steam turbines is about 5%.The exergy destruction in gas turbine is about 3%. The losses in other equipments are minimum.

5.4 INDIVIDUAL COMPONENT EXERGY EFFICINCIES

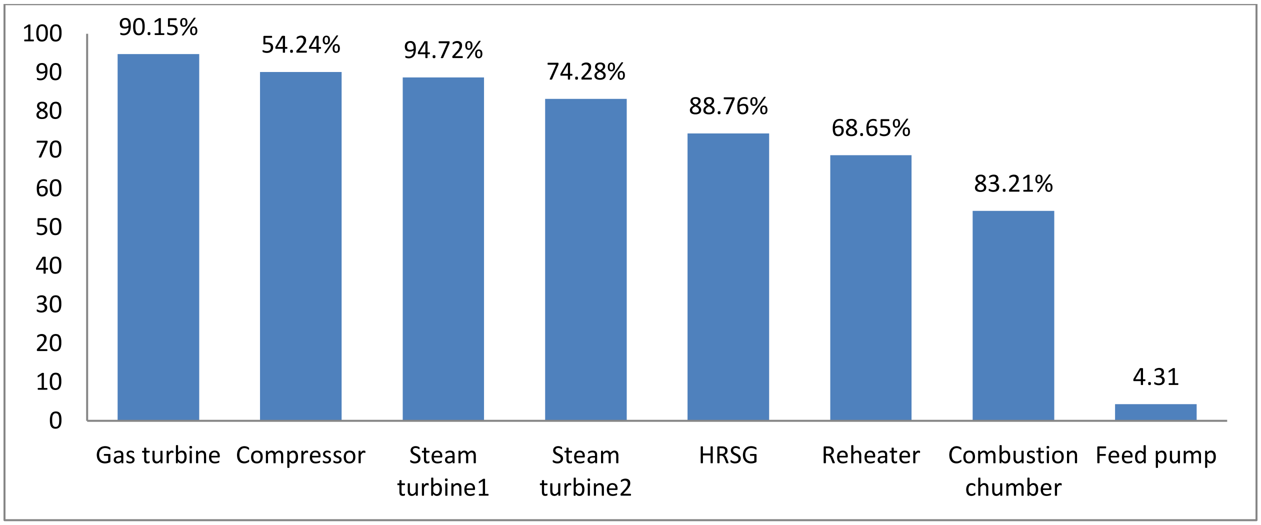


Fig 5.25 Bar graph showing Exergy efficiencies of individual components of Brayton-Rankine combined power cycle for a particular operating condition (Tmax=1373K,rp=8,Pmax=70bar,Tsteam=773K, Preheat=20 bar.Pcond=0.5 bar).

The exergy efficiency of the components are determined at a particular operating conditions as shown in fig5.24. The irreversibility's in the feed pump are very high which is reflected through its very low exergy efficiency of 4.31%. However this is irrelevant as the exergy destruction in the feed pump is very small percentage of the total exergy losses. The combustion chamber exhibits an exergy efficiency of 54%. This phenomenon can be attributed to the irreversible combustion process. The turbines have high individual exergy efficiencies in the range of 80 to 95%. The exergy losses in the heat recovery steam generator are about 25% of exergy supplied. This can be reduced by decreasing the pinch point temperature difference which will decrease the irreversibility of heat transfer process. Reheating of steam is achieved from a constant heat temperature source. Difference between the source steam temperatures was assumed as 100^0C . A higher reheater efficiency can be achieved by lowering this temperature difference.

CHAPTER 6

6 CONCLUSIONS

Based on the thermodynamic equations of the combined cycle power plant, a MATLAB program is developed and parametric analysis is carried out. The conclusions drawn from the study are enumerated as below:

- The energy and exergy efficiency of the cycle increase with the maximum temperature.
- The power output of gas turbine increases with pressure ratio while that of steam cycle decreases. Both the cycles produce equal outputs in the pressure ratio range of 3 to 3.5. The power output being higher at higher temperature for both gas and steam cycles.
- The first law efficiency of the compressor exhibits an interesting trend on the performance. For every compressor efficiency chosen, there is an optimum pressure ratio. This optimum value increases with increase compressor efficiency.
- With the increase in HRSG efficiency, both the first and second law efficiencies of the cycle increase and reach asymptotic values which decreases with decrease in HRSG efficiency.
- The effect of highest steam pressure along with pressure ratio of gas cycle reveals that

  1. low pressure ratios of gas cycles with highest steam pressures are beneficial and vice versa with regards to energy efficiency.

  2. The exergy efficiency also shows similar trends but is found to be independent of steam pressure at higher pressure ratios.

- The optimum reheat steam pressure is found to be 50% of the maximum steam pressure for the best performance.
- A drop in condenser pressure results in higher efficiency. However the enhancement of performance diminishes at higher pressure ratio.

- The maximum exergy destruction occurs in the combustion chamber and amounts to 67% of the total exergy destruction in the cycle. This is due to irreversible combustion process.

- The turbines (gas+steam) contribute to 8% of the total exergy destruction. The exergy losses in the HRSG are about 25% of exergy supplied.

- The turbines and compressor have high individual second law efficiencies of around 80% to 95%, While the feed pump exhibits very poor second law efficiency(4.31%).The combustion chamber has an exergy efficiency of 54.24%.

- Power ratio was necessary to compare the relative power outputs from gas and steam cycles. Hence, a new parameter i.e. power ratio was defined. The variation of power ratio with different parameters was studied in detail and the following were observed.

    1. The maximum temperature has an influence over power ratio as it was observed that the power ratio is higher at lower maximum temperature.
    2. The gas turbine efficiency also influences the power ratio. It has been observed that the power ratio shows a declining trend with decrease in gas turbine efficiency.
    3. The highest steam pressure of the cycle also control over the power ratio. The power ratio shows incremental tendencies with decrease in steam pressure,
    4. With the decrease in condenser pressure the power ratio enhances.

CHAPTER 7

7 REFERENCES

[1] Ali Bolattürk, and Mehmet Kanog˘lu An Investigation of Energy Versus Exergy Based Optimization for Power Cycles. Volume 24 · Number 4+5 · 2006 pp. 243–258.

[2] Tomasz Kowalczyk , Paweł Ziółkowski , and Janusz Badur , Exergy Losses in the Szewalski Binary Vapor Cycle, Entropy 2015, 17, 7242-7265; doi:10.3390/e17107242.

[3] I. G. Rice The Combined Reheat Gas Turbine/ Steam Turbine Cycle, 42 / VOL. 102, JANUARY 1980.

[4] **Harvey S. Leffa**,Thermodynamics of combined-cycle electric power plants, Am. J. Phys., Vol. 80, No. 6, June 2012.

[5] Vundela Siva Reddy, Subhash Chandra Kaushik, Sudhir Kumar Tyagi, Naraya Lal Panwar An Approach to Analyse Energy and Exergy Analysis of Thermal Power Plants

[6] A. Khaliqa, S.C. Kaushikb Second-law based thermodynamic analysis of Brayton/Rankine combined power cycle with reheat.

[7] Padma Dhar Garg, Satender Dehiya, Ankit Barasiya, Aditya Rahangdale, Vijay Shankar Kumawat, Exergy and Efficiency Analysis of Combined Cycle Power Plant.

[8] P.K Nag, “Engineering Thermodynamics” Tata McGraw HillEducation publications.

[9] Michael A.Boles and YunusA.Cengel, “Thermodynamics: An Engineering Approch”McGraw HillEducation publications.