EXERGY ANALYSIS OF DIFFERENT COMPONENTS OF STEAM POWER PLANT

A Major Thesis Submitted in Partial fulfillment of the requirements for the award of the degree of

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IN

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DECLARATION

I hereby declare that the work which being presented in the major thesis entitled **"EXERGY ANALYSIS OF DIFFERENT COMPONENTS OF STEAM POWER PLANT"** in the partial fulfillment for the award of the degree of Master of Technology in **"Thermal Engineering"** submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of **Dr. B. B. ARORA,** Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever. I confirm that I have read and understood 'Plagiarism policy of DTU'

Place: Delhi **TSEWANG TAMCHOS**

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CERTIFICATE

This is to certify that TSEWANG TAMCHOS, (Roll no. 2K15/THE/19), student of M.Tech, THERMAL ENGINEERING, Delhi Technological University, has submitted the dissertation titled **“Exergy analysis of Different Components of Steam power plant”** under our guidance towards the partial fulfillment of the requirements for the award of the degree of Master of Technology under our guidance and supervision.

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ABSTRACT

The increasing demand of power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on first law of thermodynamics only. The real useful energy loss cannot be justified by the fist law of thermodynamics, because it does not differentiate between the quality and quantity of energy. Energy analysis presents only quantities result while exergy analysis presents qualitative results about actual energy consumption. In this analysis shows exergy efficiency is less at each and every point of unit equipment's. Also presents major losses of available energy at combustor, superheater, economiser and air-pre heater section. This article shown energy, exergy efficiency, exergy destruction and energy losses comparison charts. The primary objective of this work is to analyse the system components separately and to identify and quantify the sites having largest energy and exergy losses at different load.

A numerical code is established using EES software to perform the calculations required for the thermal and exergy plant analysis considering real variation ranges of the main operating para-
meters such as pressure, temperature and mass flow rate. The effects of theses parameters on the system performances are investigated.

CONTENTS

List of Figures

Nomenclature

X	Exergy	kJ
I	Exergy destruction	kJ
X_P	Exergy Production	kJ
h	Specific Enthalpy	kJ/kg
s	Specific Entropy	kJ/kg.K
LCV	Low Calorific Value	kJ/kg
$\dot{m}_a$	Mass Flow Rate of air	kg/s
$\dot{m}_f$	Mass Flow Rate of fuel	kg/s
P	Pressure	bar
T	Temperature	K
S_{gen}	Entropy Generation	kJ/K
v	Specific Volume	m^3/kg
V	Velocity	m/s
W_C	Compressor Work rate	kW
W_p	Pump Work rate	kW
W_{ST}	Steam Turbine Work rate	kW
w	specific work output	kJ/kg
r_p	Pressure Ratio	-
$\dot{Q}$	Heat Transfer Rate	kW
q	specific heat supplied or rejected	kJ/kg
g	gravitational acceleration	m/s^2

Greek Symbols

γ	heat capacity ratio	-

Ψ	specific exergy	kJ/kg
η	efficiency	-
ρ	density	Kg/m^3

Subscripts

CV	Control Volume
out	Outlet
f	Fuel
i,in	Inlet
net	Net
e,out	Outlet
PH	Physical
sys	System
th	Thermal
Dest.	Destruction
Comb.	Combustion Chamber
Cond.	Condenser
ST	Steam Turbine
AFR	Air Fuel Ratio

CHAPTER-1

INTRODUCTION

1.1 Introduction:

Energy consumption is one of the most important indicator showing the development stages of countries and living standards of communities. Populace increase, urbanization, industrializing, and technologic advancement result specifically in expanding vitality utilization. This fast developing pattern achieves the vital natural issues, for example, contamination and greenhouse effect. Right now, 80% of power on the planet is roughly created from non-renewable energy sources (coal, oil, fuel-oil, gaseous petrol) fired thermal power plants, while the remaining of the power is repaid from various sources, for example, pressure driven, atomic, wind, solar based, geothermal and biogas. During the past several decades, the improvements in thermodynamic process of power plan mainly depends on energy analysis which is based on first law of thermodynamic. The energy analysis has some limitation like not shows the degradation of the energy quality and it also doesn't show the irreversibility of processes in the system. The energy loss cannot be shown by energy analysis because it cannot differentiate between the quality and quantity of energy.

For achieving higher efficiency an exergy analysis has to be done. So now a day, the exergy analysis which is based on the second law of thermodynamics has found as valuable technique in the assessment and improvement of thermal power plants. It shows theoretical useful amount energy available and destruction in exergy is called as Anergy. Exergy analysis assessing the source of thermodynamic inefficiencies in a thermal system. Hence, one might say that performing energetic and exergetic analysis together can give an entire delineation of system attributes. Such a far reaching examination will be a more advantageous approach for the evaluation and determination of the steps towards improvement.

In the literature, there exist a number of papers concerning energetic and exergetic performances of coal-fired thermal power plants. For example, Hasan[1] introduced thermodynamic inefficiencies and in addition sensible examination of each plant to others are identified and talked about for the coal-fired thermal power plants in Turkey. Aljundi [2] determined the performance of the plant was evaluated by a part savvy displaying and a point by point separation of vitality and exergy losses for the considered steam influence plant in Jordan. Keeping in view the fact expressed above, it can be normal that playing out an examination in light of a similar definition of execution criteria will be significant for execution correlations, appraisals and improvements for thermal power plants.

1.2 The Aim of the Project:

The primary objective of this work is to analyses the system component separately and to identify and quantify the sites having largest energy and exergy losses for the existing coal fired steam power plant in order to identify the needed improvement. With the help of this method, it would be possible to provide information about the exergy destruction and exergy losses along with their location and predict the highest exergy destructor components of the system. It will also help in suggest ways of improving the exergetic efficiency.

1.3 Outline of the Thesis:

The thesis is divided into seven chapters. In this chapter, the importance of the energy and exergy analysis of steam power plant and the objective of the study are briefly discussed.

Chapter 2 presents an literature review of covering topics related to this study, about energy analysis and exergy analysis of thermal power plant.

Chapter 3 is an overview of steam power plant system. The main emphasis is given to Rankine cycle thermodynamic and its main components.

Chapter 4 deals with the theoretical background of energy analysis, and exergy analysis, with the expressions and equations used in the mathematical model.

Chapter 5 is the base plants modeling chapter and it contains a detailed description of the plant, and energy and exergy analysis. It presents the mathematical procedure of the solution for the optimization problem.

Chapter 6 presents the results of each step of the methodology. In addition, it discusses the effect of the operating parameter on the performance parameters.

Chapter 7 concludes the study results.

CHAPTER-2
LITERATURE REVIEW

2.1 Literature:

Rashad A. [4] The performance of the plant was estimated by a component-wise modeling and a detailed break-up of energy and exergy losses for the considered plant has been presented at different loads (Maximum load, 75% load and, 50 % load). The exergy analysis of the plant showed that lost energy in the condenser is thermodynamically insignificant due to its low quality. In terms of exergy destruction, the major loss was found in the turbine where 46.1%, 59.6% and 42% of the fuel exergy input to the cycle was destroyed at50%, 75%, and full load respectively.

Oyedepo S.O. [5] Performance evaluation and economic analysis of a gas turbine power plant in Nigeria have been carried out for the period 2001–2010.It has nine gas turbine units with total capacity of 301 MW (9 * 31.5 MW). In this study emphasis has been on key performance indices (plant capacity, capacity factor, plant use factor, load factor, and utilization factor), reliability indices and cost of power outage. The study revealed that 64.3% of the plant installed capacity was available between 2001 and 2010. Also the percentage shortfall of energy generated within the period under review ranges from 4.18% to 14.53% as against the average acceptable value between 5% and 10%.

Kiran Bala Sachdeva and Karun [6] – are determined that the magnitude, location and source of thermodynamic inefficiencies of thermal power plant. The first law of thermodynamics introduces the concept of energy conservation, which states that energy entering a thermal system with fuel, electricity, flowing streams of matter, and so on is conserved and cannot be destroyed. Exergy is a measure of the quality or grade of energy and it can be destroyed in the thermal system. The second law states that part of the exergy entering a thermal system with fuel, electricity, flowing streams of matter, and so on is destroyed within the system due to irreversibility.

. **Hasan Huseyin Erdem et al. [7]** have analyzed comparatively the performance of nine thermal power plants in Turkey, from energetic and exergetic viewpoint. The power plants considered are mostly conventional reheat steam power plant fed by low quality coal.

Thermodynamic models of the plants were developed, energetic simulation results of the developed models were compared with the design values of the power plants and design point performance analyses based on energetic and exergetic performance criteria such as thermal efficiency, exergy efficiency, exergy loss, exergetic performance coefficient.

Reddy V Siva[8] have reviewed on energy analysis and exergy analysis of thermal power plants. They reviewed a thermodynamic analysis of a coal based thermal power plant and gas based cogeneration power plant in terms of energy and exergy analysis for the different components of the power plants in subcritical range. They concluded that, the major energy loss was found to occur in condenser. The exergy analysis showed that combustion chamber in both steam and gas turbine thermal power plants is main source of Irreversibility. The Irreversibility in condenser is insignificant as the low quality energy is lost in the condenser. An Exergy method of optimization gives logical solution improving the power production opportunities in thermal power plants.

Khan T.A., Khaliq A [9] described the second-law assessment of regenerative- reheat coal-fired electricity generation plant in terms of irreversibility analysis. He reported reduction in irreversible losses with the addition of backward, cascade type feedwater heater. He concluded that, incorporating reheating in a regenerative steam power cycle in subcritical range can further improve its efficiency and the total irreversible losses in the plant. These improvements become slower as the number of feed water heaters increase. The reduction in the total irreversible rate due to backward cascade feedwater heating is nearly 18%, which correspond to a 12% improvement in thermal efficiency. These estimates were increased to 24% and 14% respectively, with incorporation of reheat in addition to feedwater heating. The second-law indicates that maximum 24 exergy is destroyed in the boiler and these thermodynamic losses are significantly reduced by the incorporation of feedwater heating. The thermodynamic deviations resulting in non-ideal or irreversible functioning of various steam power plant components have been identified by Hermann. He concluded that known exergy reservoirs and flows within our sphere of influence are more than enough to provide energy services for the increasing population and activity of humankind.

Srinivas T., Gupta A.V.S.S.K.S., Reddy B.V [10] have carried out thermodynamic analysis of Rankine cycle with generalization of feed water heaters in subcritical range. They studied the effect of number of feed water heaters and bled temperature ratio on overall performance of the Rankine cycle in subcritical range. They have developed computer code for the evaluation of first law efficiency, irreversibilities and second law efficiency of Rankine cycle with different number of feed water heaters. They concluded that, greatest increment in efficiency is brought by the first heater; the increments for each additional heater thereafter successively diminish. An increase in feed water temperature reduces the heat 20 absorption from the outgoing flue gases in the economiser and may cause a reduction in boiler efficiency.

Mali Sanjay D et al [11] shows the analysis method in exergy and energy manner for thermal power plant and analysis carried out on 125MW coal based thermal power plant. The increasing demand of power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on first law of thermodynamics only. The real useful energy loss cannot be justified by the fist law of thermodynamics, because it does not differentiate between the quality and quantity of energy. Energy analysis presents only quantities results while exergy analysis presents qualitative results about actual energy consumption. In this analysis shows exergy efficiency is less at each and every point of unit equipment’s. Also presents major losses of available energy at combustor, superheater, economiser and air-pre heater section. In this article also shown energy exergy efficiency, exergy destruction and energy losses comparison charts.

A.Zaleta-Aguilar [12] Shows the analysis method of 300 MW conventional Power Plant Malfunction cost of Nozzle-blade section of steam turbine. Losses in in steam turbine stage (nozzle bucket) Malfunctions such as roughness, seal &leak clearance, erosion & sedimentation. Energy cost and economic cost of local products and fuels in each plant component. Maximum economic losses are caused by the incement of the nozzle area and the roughness of the nozzle surface. Exergy cost and economic cost of other plant components (of a power plant). General data of power plant analysed is: Steam Turbine: Nominal Power =300MW, SteamTemp SH/RH =538C/538C, Extraction = 7,steampr SH=169kg/cm2 abs. Steam flow =948420Kg/h Boiler : Type =Pressurised run on oil and gas having bioler exit pr = 175 kg/cm2.Changes in entropy with respect to enthalpy were used to find the malfunction cost of the nozzle-blade section of the turbine.

Verkhivker G.P. [13] In this study the exergy balance of a conventional Rankine steam cycle with power output of 232.6 MW. Water enters the heater sat 344 K and leaves at 453 K. The mass flow rate is G ˆ 400 kg/s. The quantitative analysis of exergy destruction showed that the principal irreversibilities are associated with the chemical transformation of exergy into heat, the subsequent transfer of this heat to the working fluid and the heat exchanger at the heater. It is shown that reduction in excergy destruction is achieved by increasing the values of the thermodynamic parameters of the working fluid supplied to the turbine and by reducing the temperature diferences of the net heaters.

Rosen Marc A. [14] The coal-fired Nanticoke Generating station (NGS)- 500 Mwe and the Pickering Nuclear Generating Station (PNGS) -500 Mwe is used for energy and exergy. analyses. In NGS, eight pulverized-coal-fired natural circulation steam generators each produce453.6 $kg \cdot s^{-1}$ steam at 16.89 MPa and 538 ◦ C. The flow rate of pressurized heavy water (D_2O) in the PTHL is 7724 $kg \cdot s^{-1}$. The D_2O is heated from 249 ◦ C and 9.54 MPa to 293 ◦ C and 8.82 MPa in the nuclear reactor

Athough energy and exergy efficiencies are the same for PNGS and similar for NGS, energy analyses do not systematically identify the location and cause of process inefficiencies, and exergy analyses do that is, energy losses are associated with emissions and exergy losses primarily with consumptions and little with cooling water and stack gases.

Lozano M.A. [15] The theoretical basis and several applications of the theory of exergetic cost, a major approach to the field of thermoeconomics, are presented. The use of the second law of thermodynamics through a systematic use of the exergy concept, the fuel-product concept based on the productive purpose of a component within an energy. The bases of thermoeconomics have to be found in the consequences which the second law imposes on the consumption of resources of every productive system.

Heberle Florian. [16] The concepts of series and parallel circuit of Organic Rankine Cycle ORC and heat production for geothermal resources with temperatures below 450 K have been studied by a second law analysis. Starting from power generation, the working fluid isopentane, isobutane, R245fa and R227ea were investigated for the use in ORC. Compared to a power generation CHP lead to an increase in the efficiency of up to 20%. The most efficient concept of CHP is the series circuit of ORC and heat generation. Fluids with low critical temperatures, such as isobutane and R227ea, should be favored in parallel circuits and power generation.

Aljundi Isam H. [17] – is studied that the exergy analysis of Al-Hussein power plant (396MW) in Jordan is presented. The performance of the plant was estimated by a component wise modeling and a detailed break-up of energy and exergy losses for the considered plant has been presented. It was found that the exergy destruction rate of the boiler is dominant over all other irreversibility in the cycle. It. Exergy analysis provides the tool for a clear distinction between energy losses to the environment and internal irreversibility in the process .

Osueke Christian O. [18] Energy and Exergy analysis of a 75MW Sapele steam power plant in Nigeria. Mass balance, energy balance and second law of efficiency has been used to calculate the exergy efficiency and destruction of each components in the power plant using different environment temperature such as 283K, 288K, 293K,298K, and 303K. Maximum energy loss occurred at the boiler where by 87.3% was destroyed, thereby reducing the rate of steam generated in the boiler and due to which there is decrease in the work done and efficiency of the steam power plant.

2.2 Research gap:

Loss value associated with the exergy destruction need to be analysed so that cost of exergy can be formulated. Detailed analysis have been done till today to show the energy analysis, exergy analysis and associated cost by different approach

CHAPTER-3

INTRODUCTION TO STEAM POWER PLANT

The steam power plant is one of the important source for the production the electricity. A steam power plant continuously converts the energy stored in the fossil fuels (i.e. coal, oil, natural gas) into the shaft work and ultimately into the electricity. A fossil fueled power plant as a bulk converter from fuel to electricity using water as the working medium. In this chapter, we will study about the plant layout, working cycle, advantages and disadvantages of steam power plant.

3. Steam Power Plant Layout and Operation

3.1. Coal fired steam Power Plant Layout :

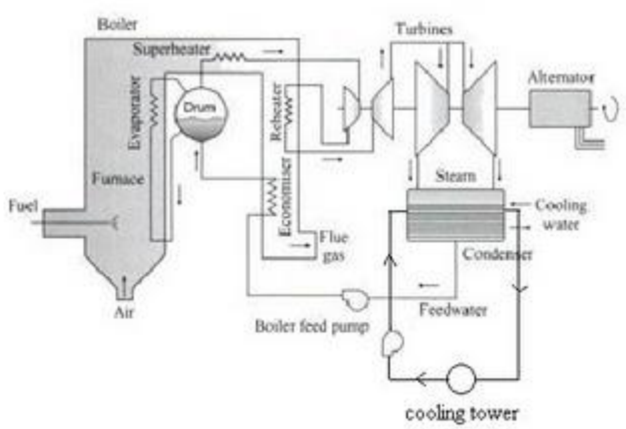

Fig 3.1 The layout of a simplified thermal power plant.

Basic Operation: A thermal power plant basically works on **Rankine cycle**.

Main parts of the plant are:

3.1.1 Coal conveyor:

This is belt kind of plan. This is used for the transportation of coal from the store to the power plant.

3.1.2 Stoker: This is device which is used to feed the coal into a combustion chamber of the furnace.

3.1.3 Pulverizer:

Before the coal are feed into the furnace they are broken into small parts or grinded. This is done by the use of the device which is called pulverizer.

3.1.4 Boiler:

Boiler is an device to produce steam. The mixture of coal and air is put into the boiler and the combustion takes place. The energy due to combustion of coal is utilizes to change the water into a steam at desired pressure and temperature. This steam then utilized in steam turbine or engine to produce electricity.

Classification of Boilers

Bolilers are classified as:

Fire tube boilers:

Hot flue gases flow into the tubes and water surrounds it. This type of boiler are usually operated at lower pressure that's is upto 20-25 bar. The rate of generation of steam is lower as compare to water tube boiler. This also requires large area for the same output steam. It is simple in design and a unskilled worker can also operate this easily. The efficiency of this kind of boiler are about 75%

Examples are: Simple vertical boiler, Cochran boiler etc.

Water tube boilers: in this kind of boiler water flows through the tube and hot gases surrounds them and heat the water to form steam. A high pressure of about 240 bar is required. The rate of steam generation is good and of good quality which can be used in power plant. The efficiency of this kind of boiler is more than 85% .

Example are: Babcock and Wilcox boiler, Stirling boiler etc.

3.1.5 Boiler feed pump:

As the water in boiler is changing into steam we need a continues supply of water. So this is a device which is responsible for the continuous supply of water to the boiler. Some water get loss in the cycle so we provide external supply of water.

3.1.6 Superheater:

Superheater is generally an integral part of a boiler boiler. Its is an important device in which a steam is heated above the saturation temperature. Its purpose is to remove the last trace of moisture from the steam. It improves the thermal efficiency of power plant.

3.1.7 Reheater :

Reheater is a kind of superheater which which increases the temperature or reheat the steam coming from the high pressure turbine before it goes to the medium and lower pressure turbine.

3.1.8 Condenser:

Condenser is a device which change the exhaust steam from the turbine to water which is than feed back to the boiler via boiler feed pump.

Condensers are classified as

(1) Jet condensers or mixed type condensers.

(2) Surface condensers or non mixed type condenser.

3.1.9 Cooling Towers:

Cooling tower is a type of heat exchanger which extract waste heat to the atmosphere by allowing the water and air to come in contact to reduce the temperature of hot water**.** Hot water is spread from top of the tower and at the same time air enters the cooling tower and start moving upward. This hot water fall down and is broken in small droplet of water. Water cools as it descends by exchanging the heat with the air. The temperature of the air increases and gets exhaust from the top of the cooling tower. The cooled water is than pumped to the condense.

3.1.10 Economiser:

An economiser is a heat exchanger, the exhaust flue gases carry a lot of heat this device utilizes the those heat increase the temperature of input water supply to the boiler. This help in increasing the efficiency of power plant. Economiser reduce the fuel consumption and there for improves the economy of the steam boiler.

3.1.11 Air preheater:

An air preheater is mainly utilized to transfer the heat energy from the exhaust flue gases to air which enters the combustion chamber of the furnace. It is installed near the boiler so that there is less loses of recover heat. This device help in reducing the fuel consumption in the furnace. This are placed b/w the economiser and the chimney.

3.1.12 Electrostatic precipitator:

It is a device which is widely used to removes dust particle from the exhaust of the power plant. In this technique the dust particle is removed by charging the particles electrically and separates them from the air under the influence of electrostatic field.

3.1.13 Generator:

The main purpose of this device is to convert mechanical energy into an electrical energy. It works on the principle of Faraday's electromagnetic induction.

3.1.14 Steam Turbine:

Steam turbine is a great prime mover to change over heat energy of steam to mechanical energy. It is one of the notable prime movers, for example, gasoline engines, diesel engines, gas turbines, jet engines, and so forth. All steam engines, regardless of whether turbines or not, are intended to concentrate energy from high-pressure steam and change it into movement by enabling the steam to extend. For the turbine designs, steam permitted to extend slowly through more than one arrangement of cutting edges, for accomplishing substantially higher efficiencies contrasted with a solitary stride extension. The steam grows through progressive rings of moving blades on a pole and fixed blades in the casing, creating purely turning movement. At the point when combined with an electric generator, steam turbine is a standout amongst the most critical methods for creating mass electric power on the planet. The advanced steam turbine may have three stages. The high-pressure segment has small blades. They are little on the grounds that the approaching steam has high energy at high temperature. After the steam goes through the high-pressure area, it is sent back to the heater to be reheated. The steam is then sent to the following area of the turbine, called the middle-pressure section. The blades here are bigger than those in the high-pressure section. In the wake of going through this segment, the steam is sent to the low-pressure area of the turbine. Since the greater part of the energy was beforehand expelled from the steam, the blades here are the biggest in the turbine. The steam leaves the turbine through the base, where it is consolidated once again into the water. From that point it is sent back to the boiler, to be made into steam once more. The steam turbine is regularly utilized as a part of a combined heat and power generation handle where the turbine drives a machine in the meantime: steam separated from the machine is utilized to supply locale heating as well as process steam systems.

A single unit of steam turbine can create power extending from 1 MW to 1000 MW. The thermal efficiencies of the present day steam power plant over 120 MW are as high as 38% to 40%. Water (steam) is the working liquid for most vapour power cycles. Water works over a wide scope of temperatures and pressures have an expansive heat limit, and it is steady, safe, and ecologically cordial. The energy sources used to produce steam incorporate gas, coal, oil, and nuclear sources.

3.2. The Ideal Rankine Cycle:

The theoretical basic cycle for the simple steam turbine power plant is the Rankine cycle which is closed one. The modern power plant uses the rankine cycle, modified to including super-heating, regenerative feed water heating and re-heating. As shown schematically a simple Rankine cycle in Fig 3.2 and T-s diagram in Fig 3.3. This is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibility and consist of the following four processes.

1-2 Isentropic compression in a pump.

2-3 Constant pressure heat addition in a boiler.

3-4 Isentropic expansion in a turbine.

4-1 Constant pressure heat rejection in a condenser.

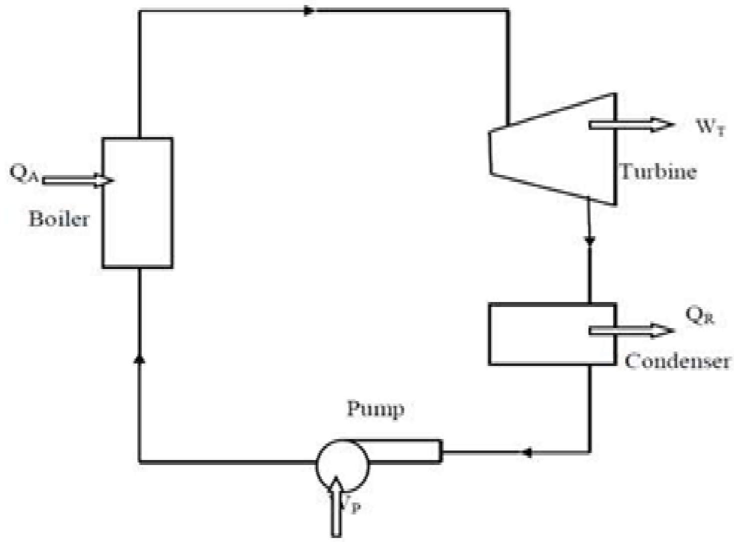

Fig 3.2 : Simple Rankine Cycle

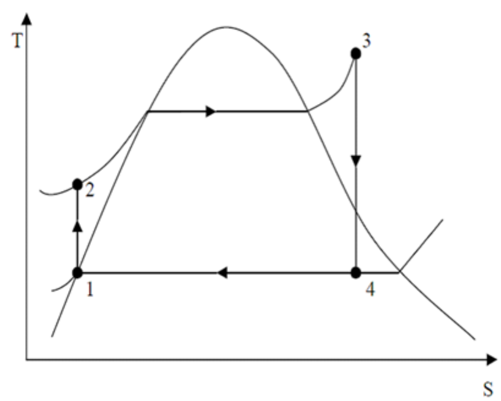

Fig 3.3: T-S Diagram for simple rankine cycle

Water enters the pump at state 1 as saturated liquid and is compressed isentropically to the operating pressure of the boiler. The water temperature increases somewhat during this isentropic compression process due to a slight decrease in the specific volume of water. The vertical distance between 1 and 2 on the T-s diagram is greatly exaggerated for clarity. Water enters the boiler as a compressed liquid at state 2 and leaves as a superheated vapor at state 3. The boiler is basically a large heat exchanger where the originating from combustion gases, nuclear reactors or other sources is transferred to the water essentially at constant pressure. The boiler, together with the section where the steam is superheated, is often called the steam generator. The superheated vapor at state 3 enters the turbine, where it expands at isentropically and produces work by rotating the shaft connected to an electric generator .The pressure and the temperature of steam drops during this process to the values at state 4, where steam enters the condenser. At this state, steam is usually a saturated liquid vapor mixture with a high quality. Steam is condensed at constant pressure in the condenser, which is basically a large heat exchanger, by rejecting heat to a cooling medium such as a cooling tower, lake, a river, or the atmosphere. Steam leaves the condenser as a saturated liquid and enters the pump, completing the cycle. Remembering that the area under the process curve on a T-s diagram represents the heat transfer for internally reversible processes, we see that the area under process curve 2-3 represents the heat transferred to the water in the boiler and the area under the process curve 4-1 represents the heat rejected in the condenser. The difference between these two is the net work produced during the cycle.

3.2.1. Deviation of Actual Vapor Power Cycles from Idealized Ones:

The actual vapor power cycle differs from the ideal Rankine cycle, as illustrated in Fig, as a result of irreversibility's in various components. Fluid friction and heat loss to the surrounding are the two common sources of irreversibility's. Fluid friction causes pressure drops in the boiler, the condenser, and the piping between various components. As a result steam leaves the boiler at a somewhat lower pressure. Also the pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the pressure drop in the connecting pipes. The pressure drop in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle calls for. This requires a larger pump and larger work input to the pump. The other major source of irreversibility is the heat loss from the steam to the surroundings as the steam flows through various components.

To maintain the same level of net work output, more heat needs to be transferred to the steam in the boiler to compensate for these undesired heat losses. As a result cycle efficiency decreases. Of particular importance is the irreversibility occurring within the pump and the turbine. A pump requires a greater work input, and a turbine produces a smaller work output as result of irreversibility's.

Under perfect conditions, the course through these gadgets is isentropic. The deviation of real pumps and turbines from the isentropic ones can be precisely represented. Nonetheless, by using isentropic efficiencies, where state 2a and 4a are the genuine leave conditions of the pump and the turbine, separately, and 4s are the comparing states for the isentropic case.

Different variables likewise should be considered in the examination of actual vapor power cycles

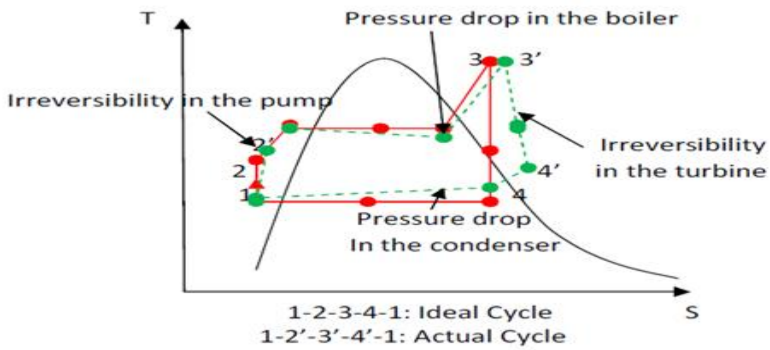

Fig 3.4 : Deviation of Rankine cycle

Extra losses happen at the direction between the moving parts thus of friction. Steam that holes out amid the cycle and air that breaks into the condenser speak to two different sources of losses. At last, the power devoured by the assistant hardware, for example, fans that supply air to the heater ought to likewise be considered in assessing the execution of actual power plants.

3.2.3. The Ideal Regenerative Rankine Cycle:

A watchful examination of the T-s chart of the Rankine cycle redrawn in Fig 3.5 uncovers that heat is exchanged to the working fluid process 2-2' at a generally low temperature. This brings down the normal heat expansion temperature and along these lines the cycle productivity. To cure this inadequacy, we search for approaches to raise the temperature of the fluid leaving the pump (called the feed water) from the extending One such plausibility is to exchange heat to the sustain water from the growing steam in a counter stream heat exchanger incorporated with the turbine, that is to utilize recovery. This arrangement is likewise unreasonable in light of the fact that it is hard to outline such a heat exchanger and on the grounds that it would expand the dampness substance of the steam at the last phases of the turbine. A pragmatic recovery prepare in steam control plants is expert by removing or "bleeding," steam from the turbine at different focuses. This steam which could have delivered more work by extending further in the turbine, is utilized to heat the bolster water rather The gadget where the nourish water is warmed by recovery is known as a regenerator, or an encourage water radiator. Recovery enhances cycle efficiency, as well as gives an advantageous method for deaerating the sustain water (evacuating the air that holes in at the condenser) to forestall erosion in the boiler. It additionally controls the expansive volume stream rate of the steam at definite phases of the turbine (because of the huge particular volumes at low weights). Subsequently regeneration is utilized as a part of all present day steam control plants. A sustain water radiator is fundamentally a heat exchanger where heat is exchanged from the steam to the feed water either by blending the two liquid streams (open feed water heater) or without blending they (shut encourage water radiators).

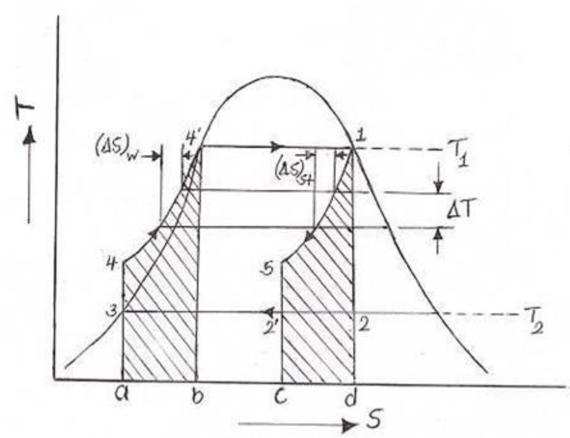

Fig 3.5: Ideal Regenerative Rankine Cycle

3.3. Rankine cycle analysis:

Rankine cycle analysis is important to understand:

3.3.1: Mass Flow Rate of the Rankine Cycle:
The mass flow rate can be measured as:

$$\dot{m}_{water} = q_{water}\rho_{water} = \frac{V_{water}}{time}\rho_{water}$$

time is calculated by a chronometer by using the volume of water available V_{water} in the boiler.

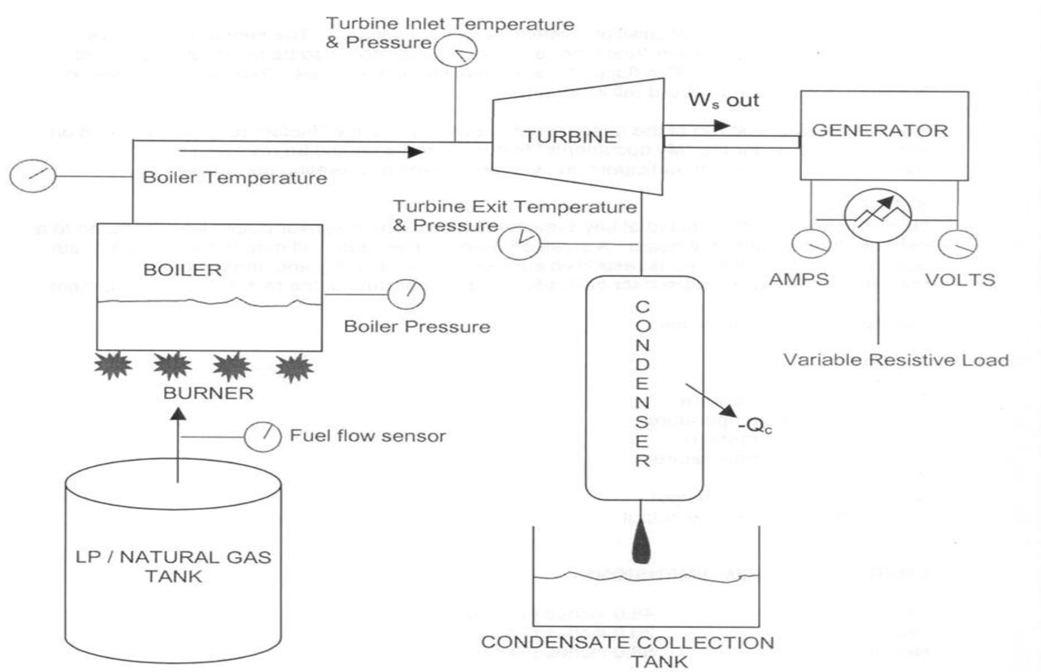

Fig 3.6 : Schematic of Rankine cycle

3.3.2 Heat Transfer and work:

For this examination, it is expected that the procedure is ideal and there are not pressure losses happening, but rather as has been said beforehand the evaporator produces superheated vapor and there is a throttling procedure in the valve.

The evaporator, for this situation a fire-tube heater, delivers a superheated vapor (Stage $1'$). Taking a control volume encasing the boiler tubes and drums, the enargy rate adjust gives:

$$0 = \dot{Q}_{in} + \dot{m}_{water}[h_{1'} - h_4 + \frac{v_1^2 - v_2^2}{2} + g(z_1 - z_2)]$$

neglecting K.E and P.E, the energy equation reduces as:

$$\dot{Q}_{in} = \dot{m}_{water}(h_{1'} - h_4)$$

At that point, vapors go through the valve, states1'- 1". For a control volume encasing the valve, the mass and energy rate reduce:

$$0 = \dot{Q}_v + \dot{m}_{water}[h_{1'} - h_{1''}]$$

$\dot{Q}_v$ can be neglected due to no heat transfer & work done, then lastly equation is:

$$h_{1'} = h_{1''}$$

which shows isenthalpic expansion in the valve.

For the pump and condenser also we do same analysis, the equation is:

$\dot{Q}_{out} = \dot{m}_{water}(h_2 - h_3)$ and $\dot{W}_p = \dot{m}_{water}(h_4 - h_3)$

The energy balance for the turbine is:

$$0 = \dot{Q}_{cv} - \dot{W}_t + \dot{m}_{water}[h_{1''} - h_{2''}]$$

Neglecting $\dot{Q}_{cv}$ to the surrounding, for turbine it is assumed as adiabatic and reversible, so is isentropic ($S_{2''} = S_{1''}$) and the energy equation is:

$$\dot{W}_t = \dot{m}_{water}(h_{1''} - h_{2''})$$

Then, we know that ($S_{2''} = S_{1''}$) and also $S_{f2}$and S_{g2}which could be find with the pressure and temperature at outlet of the turbine, the quality of the vapor can be find as:

$$x_2 = \frac{S_{g2} - S_{1''}}{S_{g2} - S_{f2}}$$

With x_2, the enthalpyh_2 is find as:

$$h_2 = h_{g2} - x_2(h_{g2} - h_{f2})$$

Where h_{f2} and h_{g2} are calculated with the outlet temperature.

All the following $h_{1'}$ $h_{1''}$ $S_{1'}$ $S_{1''}$ $S_{1,}$ $h_{B,}$ and S_4,can be determined from temperatures and pressures at each stage.

3.4. Advantages and Disadvantages of Thermal Power Plant:

Thermal power plants have been main source of power in the world for centuries. With my experience in Thermal and Gas power plant as a part of India's largest power generating companies the following are the Pros and Cons that come to my mind.

3.4..1. Advantages:

(i) Coal thermal power plant give the least expensive power around the world.

(ii) They shape the foundation of framework as they give stable yield and are more dependable than inexhaustible sources that have a tendency to fluctuate.

(iii)Coal/lignite/peat is promptly accessible everywhere throughout the world.

(iv)The innovation for thermal power plant is effortlessly open and dislike different sources like nuclear, solar thermal, Geothermal and so forth.

(v)Requires less land per Megawatt regarding Hydro, Solar, Wind and so forth.

(vi)Easy and simple to keep up and work.

3.4.2. Disadvantages

(i) Needs huge amount of Coal and Water so must be set a vast water sources and coal mine. Else the coordinations cost will be too high. A Power plant of 3000MW normally requires 35,000 to 45,000 tons of coal.

(ii)It is the prime supporter of CO_2 outflows everywhere throughout the world. With the Climate change a confinement is currently put on all nations in such manner.

(iii)Nearly 20% to 30% of coal is dismisses as Ash, which is a waste and should be dumped. Unutilized fiery debris is gathered in Ash dykes that frame the biggest land use in a run of the mill thermal power plant.It is also detrimental to environment.

(iv)Apart from CO_2, other unsafe gasses like NO_x and SO_x are likewise created prompt Acid Rain. In spite of the fact that with cutting edge burning and different procedures these are brought down and furthermore caught.

(v)Some fluid effluents are additionally released as the repercussion of water/concoction treatment and spillover of different oil/chemicals.

(vi)Thermal control plant set aside a considerable measure of opportunity to fire up (for the most part 4–8 hrs) and don't do well in cycling that is huge here and there in power era and generally intended for consistent load. In this way any change is power request can be impeding to it life cycle.

(vii)A substantial amount of ash remains is discharged from the Chimney and the coal tidy that prompt a high particulate matter in the encompassing zones. Despite the fact that almost 99% is caught utilizing ESP still a noteworthy sum get away.

CHAPTER-4
EXERGY ANALYSIS

4.1 Energy Analysis:

An energy investigation depends on the “first law of thermodynamics”. The first law of thermodynamics is all the more usually known as the law of energy conversion. The first law of thermodynamics demonstrates that energy can neither be made nor annihilated, and it can just change starting with one shape then onto the next. This law characterizes inner vitality as a state work, and gives a formal articulation of the conservation of energy. The most common energy systems, such as power generation and refrigeration systems, are open systems (systems in which mass flows through the various components). The typical components of power and refrigeration systems are boilers, turbines, evaporators etc., all of which have inlets and outlets.

An energy study is generally used in estimating the performance of a part or a system, and can be used to regulate the first law efficiency (η_{th}) for a power production cycle.

Be that as it may, an energy adjust gives no data about the heading in which procedures can suddenly happen or potentially the reversibility of the thermodynamic procedures. The first law can't give data about the powerlessness of any thermodynamic procedure to change over heat completely into mechanical work, or any understanding into why mixture can't suddenly isolate themselves.

4.2 Exergy Analysis:

Exergy can be defined briefly as: the maximum theoretical work obtained from a system when this system is brought from a state to equilibrium with the environment while interacting only with the environment. The state of a system is defined by temperature, pressure, and composition. Moreover, exergy is the base theoretical work expected to carry the system from equilibrium with nature to the given state. This implies exergy is a measure of the take-off of the condition of a system from the condition of the surroundings. This makes exergy a property of both the system and condition together. The meaning of exergy won't be finished, in any case, unti Exergy of a System.

The total exergy is that exergy that can be extracted through heat and work processes, hence,

$$\Delta\dot{E}_{system} = \Delta\dot{E}_{heat} - \Delta\dot{E}_{work} \quad 4.1$$

4.3 Exergy associated with work transfer: from the definition of the work, that equivalent a given type of energy as a measure of its exergy, unmistakably work is comparable to exergy in each regard. In this way, exergy transfer can be indicated both in size and in bearing by the work transfer to which it corresponds

$$\Delta\dot{E}_{work} = W + \int P_0\, dV \quad 4.2$$

4.4 Exergy associated with heat transfer: supposing a constant temperature dispersal in a thermal energy reservoir, the exergy transfer rate, $\Delta\dot{E}_{work}$ associated with the heat transfer rate Q, can be intended by the subsequent method work:

$$\Delta\dot{E}_{heat} = (1 - \frac{T_0}{T})\dot{Q} \quad 4.3$$

substituting equation (4.2) and (4.3) in equation (4.1),

$$\Delta\dot{E}_{heat} = (Q - T_0 \int dS) - (W - \int P_0 dV) \quad 4.4$$

and rearranging

$$\Delta\dot{E}_{heat} = Q - W - T_0 \int dS + \int P_0 dV \quad 4.5$$

Introducing the first law:

$$\Delta E = Q - W \quad 4.6$$

eliminates Q and W yielding,

$$\Delta\dot{E}_{system} = \Delta E - T_0 \int dS + \int P_0 dV \quad 4.7$$

expanding

$$\Delta\dot{E} = \Delta U - T_0 \int dS + \int P_0 dV + m\frac{1}{2}V^2 + mgz \quad 4.8$$

Integrating the above between the state of the system and the dead state yields,

$$\dot{E} = U - U_0 - T_0(S - S_0) + p_0(V - V_0) + m\frac{1}{2}V^2 + mgz \quad 4.9$$

Here, V is the velocity of the system relative to the reference frame of the environment and z is the height of the system relative to the reference frame of the environment (where $z_0 = 0$ usually the ground).

Exergy Component

The total exergy of a system $\dot{E}$ can be divided into four components: physical exergy $\dot{E}_{PH}$, kinetic exergy $\dot{E}_{KN}$, potential exergy $\dot{E}_{PT}$, and chemical exergy $\dot{E}_{CH}$:

$$\dot{E}=\dot{E}_{PH}+\dot{E}_{KN}+\dot{E}_{PT}+\dot{E}_{CH} \qquad 4.10$$

4.5 Potential Exergy:

Potential exergy, $\dot{E}_{PT}$ is determined with potential energy with respect to zero level associated to the surroundings. Since potential energy must consider every one of the strengths influencing the analysed matter and condition, it implies that, other than the surrounding constraint, one ought to likewise consider the drive created by the pressure of nature parts. Accepting that the speeding up of gravity does not fluctuate with height, potential energy can be figured from the accompanying:

$$\dot{E}_{PT} = m(gz - g\int_0^h \gamma_0 \, dh) \qquad 4.11$$

where g , h and γ_0 are gravity acceleration, height of matter centre with respect to zero level and environment density respectively.

4.6 Physical Exergy:

Physical exergy speaks to a piece of the exergy that shows up because of the distinction in temperature and pressure of the observed matter and the temperature and pressure of the environment(T_0, P_0). It is naturally divisible into two components.

Thermal component $E_{PH}^{\Delta T}$, is the component ensuing from the temperature change between the stream and the environment:

$$E_{PH}^{\Delta T} = mC_p\left[(T - T_0) - T_0 \ln\frac{T}{T_0}\right] \qquad 4.12$$

Pressure component $E_{PH}^{\Delta P}$, the component ensuing from the pressure change between the stream and the environment:

$$E_{PH}^{\Delta P} = (h - h_0) + T_0(s - s_0) \qquad 4.13$$

4.7 Exergy Balance:

By combining the first and second law of thermodynamics, the mathematical equation of an exergy balance is

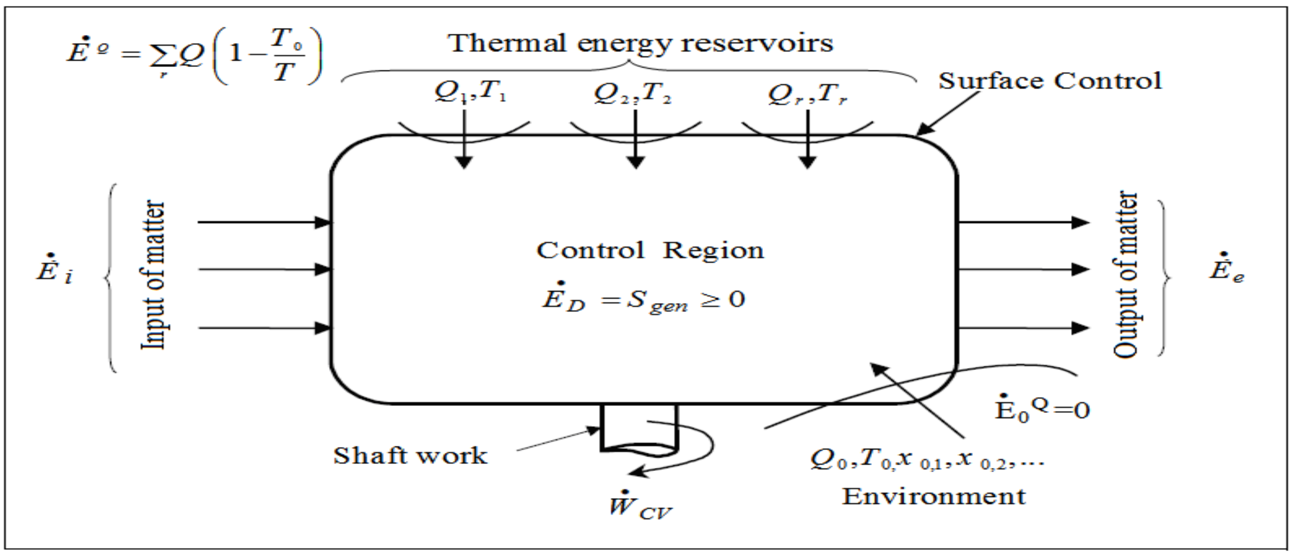

Fig 4.1: Steady state process in an open control region

An expression of an exergy analysis for an open system is: $\begin{bmatrix} Rate\ of\ exergy \\ change\ within \\ control\ volume \end{bmatrix} =$

$$\begin{bmatrix} Net\ exery \\ transfer \\ by\ heat \end{bmatrix} + \begin{bmatrix} Net\ Exergy \\ transfer \\ by\ work \end{bmatrix} - \begin{bmatrix} Rate\ of \\ Exergy\ addition \\ with\ mass \end{bmatrix} + \begin{bmatrix} Rate\ of \\ Exergy\ removal \\ with\ mass \end{bmatrix} +$$

$$\begin{bmatrix} Exergy \\ destruction \\ within \\ control\ volume \end{bmatrix} \qquad 4.14$$

$$\left(\frac{d\dot{E}}{dt}\right) = \sum_j (1 - \frac{T_0}{T_j})\dot{Q}_J\text{-}W_{CV} + \sum m_i \dot{e}_i - \sum m_e \dot{e}_e - \dot{E}_D \qquad 4.15$$

Where $\left(\frac{d\dot{E}}{dt}\right)$ is the time rate of change of the exergy stored within the control volume, $\left(1 - T0TjQj\right.$ is the net exergy change due to heat transfer, Tj is the temperature at jth control volume boundary (where the heat is crossing into the control volume at that boundary); W_{CV} is the net exergy transfer due to non-flow work out of the control volume; $\sum m\dot{e}$ is the exergy addition (i) or removal (o) due to mass flow; $\dot{E}_D = T_0 \, S_{gen}$ and is the exergy destruction within the control volume.

4.8 Exergy Wastes (Exergy Destructions and Exergy Losses):

The thermal system under consideration for analysis is supplied with some input (fuel exergy $\dot{E}_F$) derived from energy source. This input transfers into some exergy output (product exergy $\dot{E}_p$). For a real process the exergy input always exceeds the exergy output, and this unbalance is due to waste in exergy; it is useful to differentiate between types of exergy wastes in order to study where irreversibilities occur. Two kinds of exergy wastes can be distinguished: internal and external.

External exergy wastes (exergy losses $\dot{E}_{loss}$) represent the remaining exergy contents of losses and emissions that are dissipated or removed from the production and embody, thus unused (exergy remaining non-utilized output).

Internal exergy wastes (exergy destruction $\dot{E}_D$) correspond to the wastes of quality due to internal inefficiencies within the process, it is the direct result of the irreversibilities in a system. These internal irreversibilities may be of technical nature due to technical inefficiencies within the plant, e.g. friction or lack of insulation, or they may be of a structural nature. Structural exergy destruction $\dot{E}_D$ is determined by the principle and design of the system. Whereas technical exergy destruction $\dot{E}_D$ can be reduced through optimization, structural waste can be reduced only by redesigning the system.

The exergy destruction is related to the entropy generation equation.

$$\dot{E}_{loss} = \dot{E}_{out} - \dot{E}_p \quad 4.16$$

$$\dot{E}_D = \dot{E}_{in} - \dot{E}_{out} \quad 4.17$$

$$\dot{E}_{waste} = \dot{E}_D + \dot{E}_{loss} \quad 4.18$$

For the exergy analysis, it is necessary to define product and fuel for each component and for overall system. The product is defined according to the purpose of owning and operating the component under consideration and fuel represent the resources consumed in generating the product. Fuel and product are expressed in terms of exergy. Exergy destruction is the amount of exergy lost due to irreversibilities and cannot be used anywhere. The exergy losses are the amount of exergy that is lost from the system under consideration, but can be useful to other system.

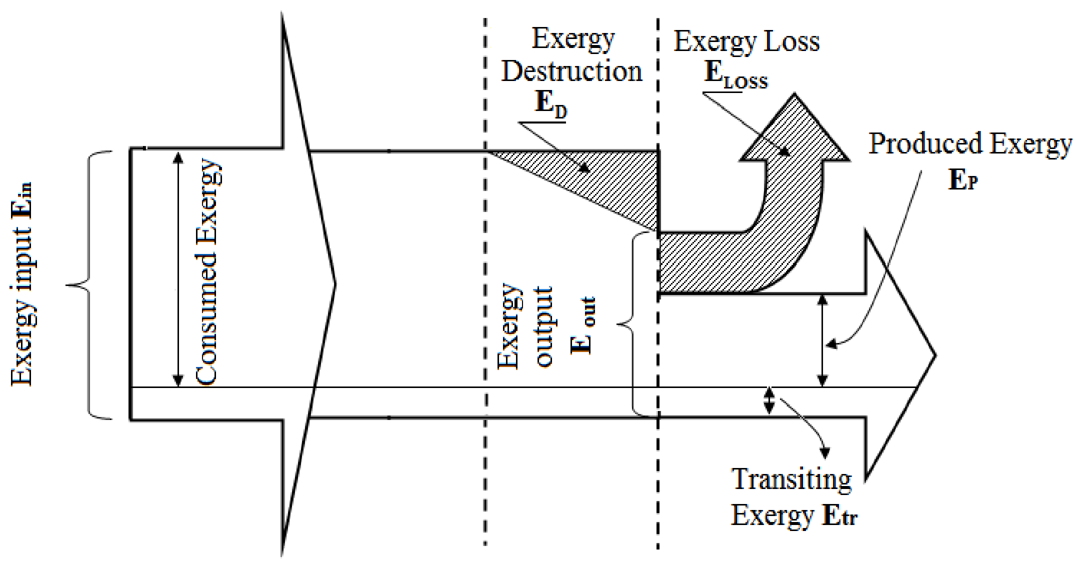

Figure 4.2: Graphical presentation of overall exergy balance

The exergy destruction can be calculated from equation (4.15), the exergy destruction equation for any control volume at steady state with insignificant kinetic and potential energy changes can be written as in equation (4.19).

$$\dot{E}_D = \sum_j (1 - \frac{T_0}{T_j})\dot{Q}_j \text{-} W_{CV} + \sum \dot{E}_i - \sum \dot{E}_o \quad 4.19$$

An exergy balance, by definition, exists for reversible procedures. Subsequently, for genuine procedures, i.e. irreversible procedures, exergy is never in balance, in light of the fact that the aggregate exergy input dependably surpasses the aggregate exergy yield, i.e. Thus, it is deceiving to discuss an exergy adjust for genuine procedures.

By ascertaining the exergy waste, i.e. pulverization and loss, we can imagine conceivable process upgrades. By and large, when the exergy loss is high, we ought to consider enhancing this part first. In any case, this approach is not generally fitting. The reason is that all aspects of the system rely on upon each other and a change in one section may increase losses in different parts so that the aggregate losses in the balanced procedure might be equivalent or significantly bigger than in the first procedure setup. Along these lines, the issue needs an all the more precisely approach, which we will talk about beneath.

giving the exergy destruction ratio:

$$y_D = \frac{\dot{E}_D}{\dot{E}_{F,TOT}} \qquad 4.20$$

$\dot{E}_{D,TOT}$ giving the ratio:

$$y_D^* = \frac{\dot{E}_D}{\dot{E}_{D,TOT}} \qquad 4.21$$

exergy of the fuel provided to the overall system:

$$y_{loss} = \frac{\dot{E}_{loss}}{\dot{E}_{F,TOT}} \qquad 4.22$$

4.9 Exergy Efficiency:

Exergy efficiencies can be used for various purposes. An obvious application is to use them for assessing, analysing, and optimizing processes and systems. Exergy efficiencies are particularly valuable in analysing and optimizing systems. An exergy investigation usually includes a comprehensive calculation of the exergy standards of process flows and the exergy waste in the system. Such a calculation shows the places in the system where waste occur. In the analysis, the question that has to be answered is how the exergy losses can be avoided or limited. Based on the absolute value of exergy loss, it is usually difficult to assess whether an exergy waste in an apparatus is unnecessarily large. An exergy efficiency in which the exergy loss is associated with the added or transferred exergy gives a better picture of the value of the processes in the apparatus, and thus gives a better impression of whether exergy waste can be reduced.

The calculation of exergy efficiency can be a rather difficult subject due to the lack of calibration and uncertainty of some terms set up in the literature.Two main classes of exergy efficiencies definitions will be presented next, universal exergy efficiency and functional exergy efficiency, and several authors have provided these definitions .

4.10 Universal Exergy Efficiency:

Universal exergy efficiency is defined as a ratio of gross exergy yield to gross exergy input Fig 4.3. There are two main modules of universal exergy efficiencies reported in the literatures. The primary is simple efficiency and the second is the efficiency with transiting exergy .

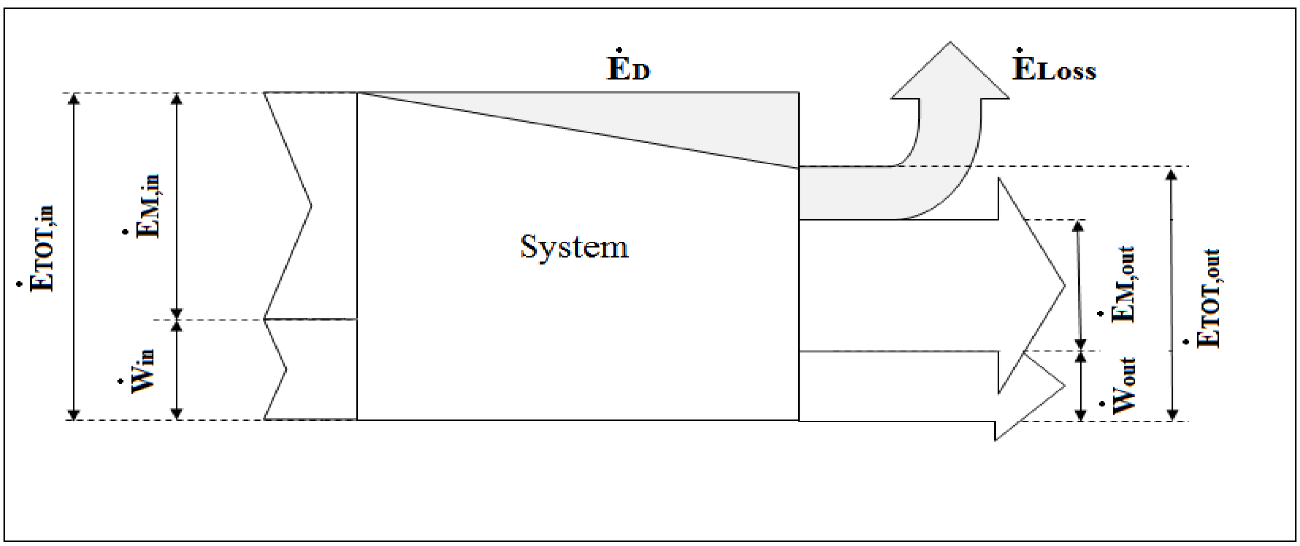

Fig 4.3: Simple exergy efficiency

Because of its simplicity, it is a generally applicable definition for exergy efficiency. This form of efficiency is based on the exergy balance to express all exergy input as used exergy, and all exergy output as utilized exergy equation (4-23). Therefore, the exergy efficiency ε becomes:

$$\boldsymbol{\varepsilon} = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \mathbf{1} - \frac{\dot{E}_D}{\dot{E}_{in}} \qquad 4.23$$

However, this efficiency does not always deliver a passable classification of the thermodynamic efficiency of processes, such as heat transfer, separation, expansion etc.

There is often a part of the output exergy which is unused, i.e. an exergy loss $\dot{E}_{loss}$ to the environment; moreover, the exergy efficiency ε_1 becomes loss:

$$\varepsilon_1 = \frac{\dot{E}_{out} - \dot{E}_{loss}}{\dot{E}_{in}} \tag{4.24}$$

- Efficiency with Transiting Exergy

Efficiency with transiting exergy ε_{tr} is seen as a development of the simple efficiency. The unreconstructed components are here subtracted from the inward and the outward-bound components. The efficiency will be defined by:

$$\varepsilon_{tr} = \frac{\dot{E}_{out} - \dot{E}_{tr}}{\dot{E}_{in} - \dot{E}_{tr}} \tag{4.25}$$

Where $\dot{E}_{tr}$ is the transiting exergy and it is defined as the part of the exergy which negotiates a system without taking any part in the mechanical, thermal or chemical changes which take place in the system.

The universal efficiency offers a reasonable definition for an assortment of frameworks. A detriment of this definition, notwithstanding, is that the efficiencies values acquired can be cold-hearted to changes in the system.

CHAPTER 5

Formulation of Equation

Table 5.1 Main parameters used of the steam power plant.

Parameters	Value
The isentropic efficiency of steam turbine	90
The isentropic efficiencies of water pump	82
Air fuel ratio	15 to 25
The inlet air pressure in the furnace	1bar to 4bar
The inlet air temperature in the furnace	303 K to 333K
Boiler pressure	10 bar and 12 bar

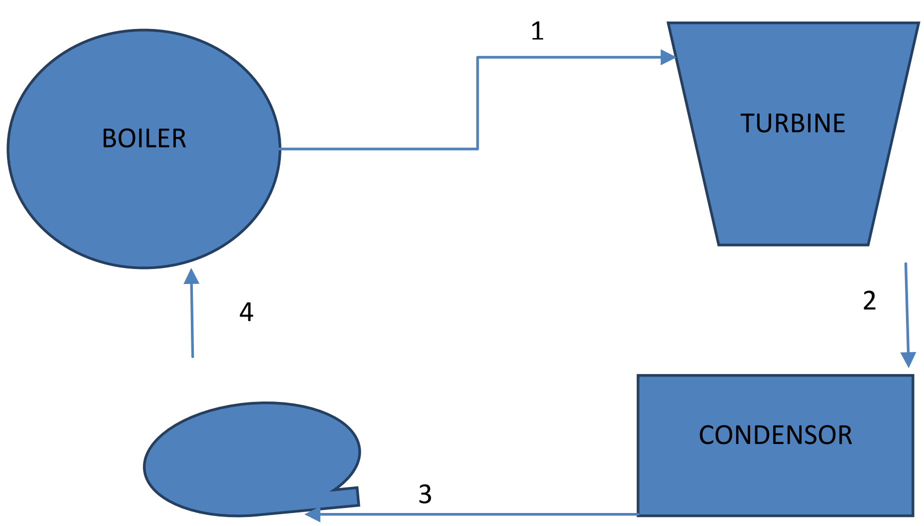

Fig 5.1 : Steam power plant cycle

Table 5.2: Table indicating various points of schematic diagram.

Point number	Specification
1	Superheated steam entering steam turbine
2	Outlet steam from steam turbine
3	Saturated liquid entering feed water pump
4	Liquid entering the boiler

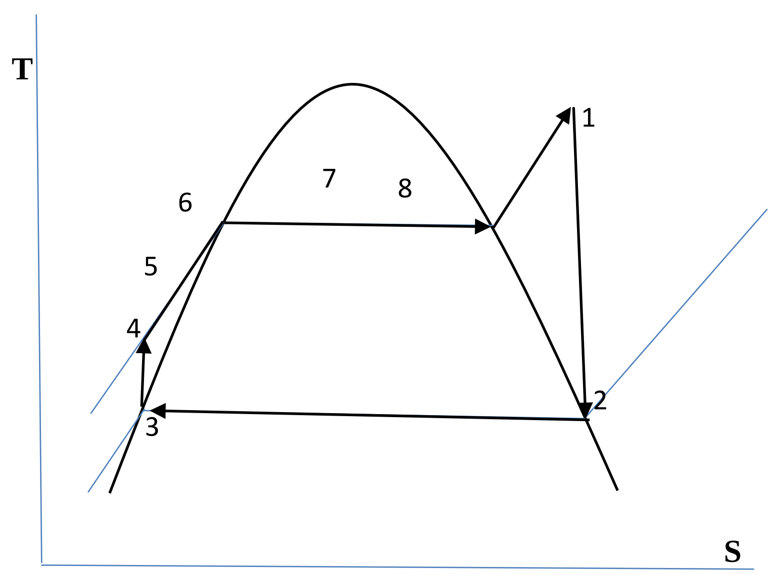

Fig.5.2 : T-S diagram of steam cycle

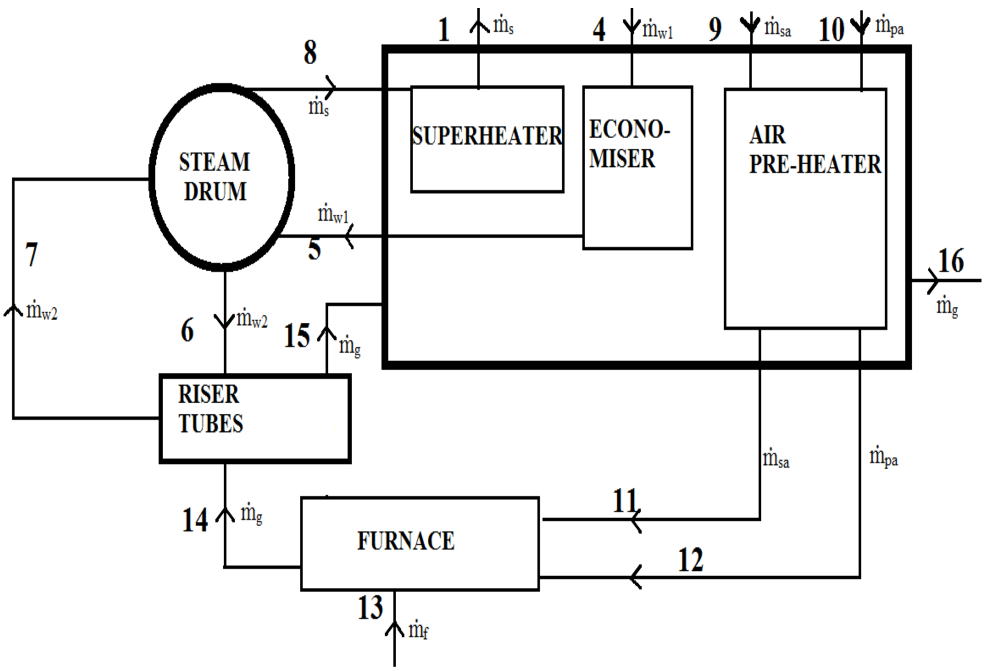

Fig. 5.3: Schematic representation of Boiler.

5.1 Energy and Exergy equations of different components steam power plant:

The basic equations employed in the exergy analysis performed on the steam cycle power plant are presented in this section. As with the energy analysis, exergy balances for individual components are written, and exergy flows and irreversibilities for each component are found. Then, overall exergy efficiency and exergy destruction are found for the whole system. Following are the equation of the first law efficiency, second law efficiency and exergy destruction in different component of steam power plant.

5.1.1 Boiler:

(a) Furnace:

Energy analysis :-

$$\boldsymbol{E_i = \dot{m}_f h_f + \dot{m}_{pa} h_{pa} + \dot{m}_{sa} h_{sa}} \quad (1)$$

$$E_o = \dot{m}_g h_g \qquad (2)$$

$$E_{loss} = E_i - E_o \qquad (3)$$

$$\eta_i = \frac{E_o}{E_i} \qquad (4)$$

Exergy analysis :-

$$X_i = \mathcal{E}_f + \mathcal{E}_{pa} + \mathcal{E}_{sa} \qquad (5)$$

$$X_o = \mathcal{E}_g \qquad (6)$$

$$I = X_i - X_o \qquad (7)$$

$$\eta_{ii} = \frac{X_o}{X_i} \qquad (9)$$

Where-

$$\mathcal{E}_{pa} = \dot{m}_{pa}(h_{pa} - T_o s_{pa}) \qquad (a)$$

$$\mathcal{E}_{sa} = \dot{m}_{sa}(h_{sa} - T_o s_{sa}) \qquad (b)$$

$$\mathcal{E}_g = \dot{m}_g(h_g - T_o s_g) \qquad (c)$$

For calculating the Exergy of fuel, it is proposed by Shieh and Fan

$$\mathcal{E}_f = 34183.16(C) + 21.95(N) + 116559.9\,(H) + 18242.90\,(S) + 13265.9\,(O)K$$

(b) Heat Recovery System (HRS):

<u>Energy analysis:-</u>

$$\boldsymbol{E_i = \dot{m}_g(\Delta h_{i-o})_g} \qquad (10)$$

$$\boldsymbol{E_o = \dot{m}_{sup}(\Delta h_{i-o})_{sup} + \dot{m}_w(\Delta h_{i-o})_w + \dot{m}_{sa}(\Delta h_{i-o})_{sa}} \qquad (11)$$

$$\boldsymbol{E_o = \dot{m}_{sup}(h_o - h_i)_{sup} + \dot{m}_w(h_o - h_i)_w + \dot{m}_{sa}(h_o - h_i)_{sa}} \qquad (12)$$

$$\boldsymbol{E_{loss} = E_i - E_o} \qquad (13)$$

$$\eta_{\boldsymbol{I}} = \frac{\boldsymbol{E_o}}{\boldsymbol{E_i}} \qquad (14)$$

<u>Exergy Analysis:-</u>

$$\boldsymbol{X_i = \mathcal{E}_{gi} - \mathcal{E}_{go} = \dot{m}_g(h_g - T_0 s_g)_i - \dot{m}_g(h_g - T_o s_g)_o} \qquad (15)$$

$$\boldsymbol{X_i = \left(\mathcal{E}_{sup(o)} - \mathcal{E}_{sup(i)}\right) + \left(\mathcal{E}_{w(o)} - \mathcal{E}_{w(i)}\right) + \left(\mathcal{E}_{pa(o)} - \mathcal{E}_{pa(i)}\right) + \left(\mathcal{E}_{sa(o)} - \right.}$$
$$\boldsymbol{\mathcal{E}sai} \qquad (16)$$

$$\boldsymbol{X_o = \left[\dot{m}_{sup}\left(h_{sup} - T_o s_{sup}\right)_o - \dot{m}_{sup}\left(h_{sup} - T_o s_{sup}\right)_i\right] + [\dot{m}_w(h_w -}$$
$$\boldsymbol{Toswo - mwhw - Toswi + mpahpa - Tospao - mpahpa - Tospai +}$$
$$\boldsymbol{[\dot{m}_{sa}(h_{sa} - T_o s_{sa})_o - \dot{m}_{sa}(h_{sa} - T_o s_{sa})_i]} \qquad (17)$$

$$I = X_i - X_o \tag{18}$$

$$\eta_{II} = \frac{X_o}{X_i} \tag{19}$$

5.1.2. Analysis of Turbine:

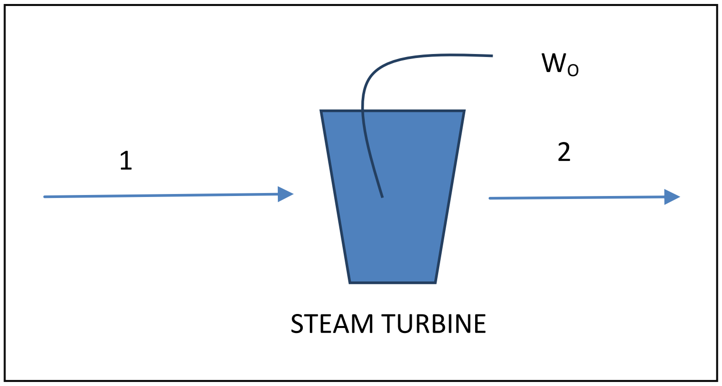

Fig.5.4: Steam Turbine

(a) Energy analysis:

For adiabatic turbine-

$$E_i = \dot{m}_s h_1 \tag{20}$$

$$E_o = \dot{m}_s h_2 \tag{21}$$

Energy balance:

$$\mathbf{E_i = E_o + W} \tag{23}$$

$$\eta_{\mathrm{I}} = \mathbf{1} - \frac{\mathbf{E_o}}{\mathbf{E_o}} \tag{24}$$

(b) Exergy analysis:

$$X_i = \dot{m}_s (h_1 - T_0 s_1) \tag{25}$$

$$X_o = \dot{m}_s (h_2 - T_0 s_2) \tag{26}$$

Exergy balance:

$$\mathbf{X_i - I - W_o = X_o} \tag{27}$$

$$\eta_{\mathrm{II}} = \mathbf{1} - \frac{\mathbf{I}}{(\mathbf{X_i - X_o})} \tag{28}$$

5.2 Equation of the exergy destruction rate and the exergy efficiency of different component of plant:

5.2.1 Exergy Destruction Of Different Component are:

(a) Pump:

$$\mathbf{I_p = X_3 + W_p - X_4} \quad (29)$$

(b) Condenser:

$$\mathbf{I_{con} = (X_2 - X_3) - (X_{3'} - X_{2'})} \quad (30)$$

(c) Turbine:

$$\mathbf{I_t = X_i - X_o - W_t} \quad (31)$$

5.2.2 Exergy Efficiency of Different Component are:

(a) Pump:

$$\boldsymbol{\eta = \frac{X_4 - X_3}{W_p}} \quad (32)$$

(b) Condenser:

$$\boldsymbol{\eta = 1 - \frac{Icon}{(X2 - X3)}} \quad (33)$$

(c) Turbine:

$$\boldsymbol{\eta = \frac{W_t}{X_i - X_o}} \quad (34)$$

CHAPTER-6
RESULT AND DISCUSSION

In this study, an energetic and exergetic analysis has been presented. The effect of varying temperature and pressure of inlet air, air fuel ratio, boiler pressure and temperature on the energy and exergy of a steam power plant has been shown. Mass balance, energy balance and second law of efficiency were used to calculate the exergy efficiency and destruction of each components in the steam power plant.

6. Result from Energy and Exergy analysis:

6.1 Effect of A/F (air fuel ratio) on Temperature of the hot gas at different inlet air pressure:

- Fig 6.1. Illustrates the Variation of hot gas temperature in furnace with A/F ratio at different air pressure. Air fuel ratio was varied from 15 to 25. Pressure was varied from 1 bar to 4 bar.
- At all the pressure value with the increase in air fuel ratio there is continuously decrease in the temperature of the hot gas in the furanace because as air fuel ratio increases the quantity of air continuously increases which support the combustion so in starting temperature of hot gases decreases slowly but as air fuel increases more the heat of the combustion absorbed by air which reduces the hot gas temperature.
- Important point to be noted is that, there is no effect on the hot gas temperature with the increase in pressure value.

6.2 Effect of A/F (air fuel ratio) on Exergy of the fuel (e_g) at different inlet air temperature:

- Fig 6.2. Illustrates the variation of Exergy of hot gas with A/F ratio at different air temperature. Air fuel ratio was varied from 15 to 25. Temperature was varied from 303 K to 333 K.

- The exergy of the fuel decreases with the increase in the air fuel ratio. So at lower air fuel ratio the is higher value of exaegy of the fuel because heat of the combustion is absorbed by the air molecule added which reduces the hot gases temperature in furnace.
- At a particular air fuel ratio, as the temperature increases exergy destruction also increases because air enter with more heat contain and brings more exergy into the furnace so the overall exergy destruction rate of hot gases in furnace increases.

6.3 Effect of A/F (air fuel ratio) on Second law efficiency of HRS at different boiler pressure P_1 (bar):

- Fig6.3. Illustrates the Variation of second law eff. of HRS with A/F ratio at different boiler Pressure. Air fuel ratio was varied from 15 to 25. Pressure was varied from 10 bar to 12 bar.
- As the air fuel ratio increase the second law efficiency of HRS increases. So at higher air fuel ratio the higher is the second law efficiency of HRS.
- But with the increase in the boiler pressure there is no noticeable change in the second law efficiency of the HRS.

6.4 Effect of A/F (air fuel ratio) on Exergy destruction in HRS at different boiler pressure P_1 (bar):

- Fig 6.4. Illustrates the Variation of Exergy destruction rate of HRS with A/F ratio at different boiler Pressure. Air fuel ratio was varied from 15 to 25. Pressure was varied from 10 bar to 12 bar.
- As the air fuel ratio increases there is an decrease in the exergy destruction rate of the HRS. At higher value of the air fuel ratio, the lower is the exergy destruction rate of HRS.
- Also with the increase in the boiler pressure at a particular air air fuel ratio, there is no noticeable change in the exergy destruction rate of HRS.

6.5 Effect of turbine inlet temperature T_1 on Work output from the turbine at different boiler pressure P_1 (bar):

- Fig 6.5. Illustrates the Variation of Work output of turbine with T_1 at different boiler Pressure. The turbine inlet temperature was varied from 570 K to 640 K. Pressure was varied from 10 bar to 15 bar.
- As turbine inlet temperature increase there is increases in the work output of the turbine because enthalpy is the function of inlet temperature. As the turbine inlet temperature increases enthalpy increase so work output of turbine increases.
- At a particular turbine inlet temperature there is increase in the work output of turbine with the increases in boiler pressure.

6.6 Effect of turbine inlet temperature T_1 on First law efficiency of turbine at different boiler pressure P_1 (bar):

- Fig 6.6. Illustrates the Variation of first law eff. of turbine with T_1 at different boiler Pressure. The turbine inlet temperature was varied from 570 K to 640 K. Boiler pressure was varied from 10 bar to 15 bar.
- As turbine inlet temperature increase there is increases in the first law efficiency of the turbine because power obtained increases more as compare to heat added.
- At a particular turbine inlet temperature, there is increase in the first law efficiency of turbine with the increases in boiler pressure.

6.7 Effect of turbine inlet temperature T_1 on Second law efficiency of turbine at different boiler pressure P_1 (bar) :

- Fig 6.7. Illustrates the Variation of second law eff. of turbine with T_1 at different boiler Pressure. The turbine inlet temperature was varied from 570 K to 640 K. Boiler pressure was varied from 10 bar to 15 bar.
- As turbine inlet temperature increase, there is increases in the second law efficiency of the turbine because exergy destruction rate decreases and power obtain increases.
- But at a particular turbine inlet temperature, there is decrease in the second law efficiency of turbine with the increases in boiler pressure

6.8 Effect of turbine inlet temperature T_1 on Exergy destruction in turbine at different boiler pressure P_1 (bar):

- Fig 6.8. Illustrates the Variation of Exergy destruction in turbine with T_1 at different boiler Pressure. The turbine inlet temperature was varied from 570 K to 640 K. Bolier pressure was varied from 10 bar to 15 bar.
- As turbine inlet temperature increase, there is decerese in the exergy destruction in the turbine becauses more useful work obtained is more from steam turbine at higher inlet temperatures .
- At a particular turbine inlet temperature, there is increases in the exergy destruction in turbine with the increases in boiler pressure.

6.9 Effect of turbine inlet temperature T_1 At on Second law efficiency of condenser different boiler pressure P_1 (bar) :

- Fig 6.9. Illustrates the Variation of Second law efficiency of condencer with T_1 at different boiler Pressure. The turbine inlet temperature was varied from 570 K to 640 K. Bolier pressure was varied from 10 bar to 15 bar.
- As TIT increase, there is increases in the second law efficiency of the condenser because the exergy entering in the condenser is increases as TIT increases.
- But at a particular turbine inlet temperature, there is decrease in the second law efficiency of turbine with the increases in boiler pressure because exergy entering in the condenses increases as boiler pressure increases because more steam expanded in steam turbine.

6.10 Effect of turbine inlet temperature T_1 on Exergy destruction in condenser at different boiler pressure P_1 (bar) :

- Fig 6.10. Illustrates the Variation of Exergy destruction in condencer with turbine inlet temperature at different boiler Pressure. The turbine inlet temperature was varied from 570 K to 640 K. Bolier pressure was varied from 10 bar to 15 bar.
- As turbine inlet temperature increase, there is decrease in the exergy destruction of the condenser.
- At a particular turbine inlet temperature, there is increase in the exergy destruction of condenser with the increases in boiler pressure

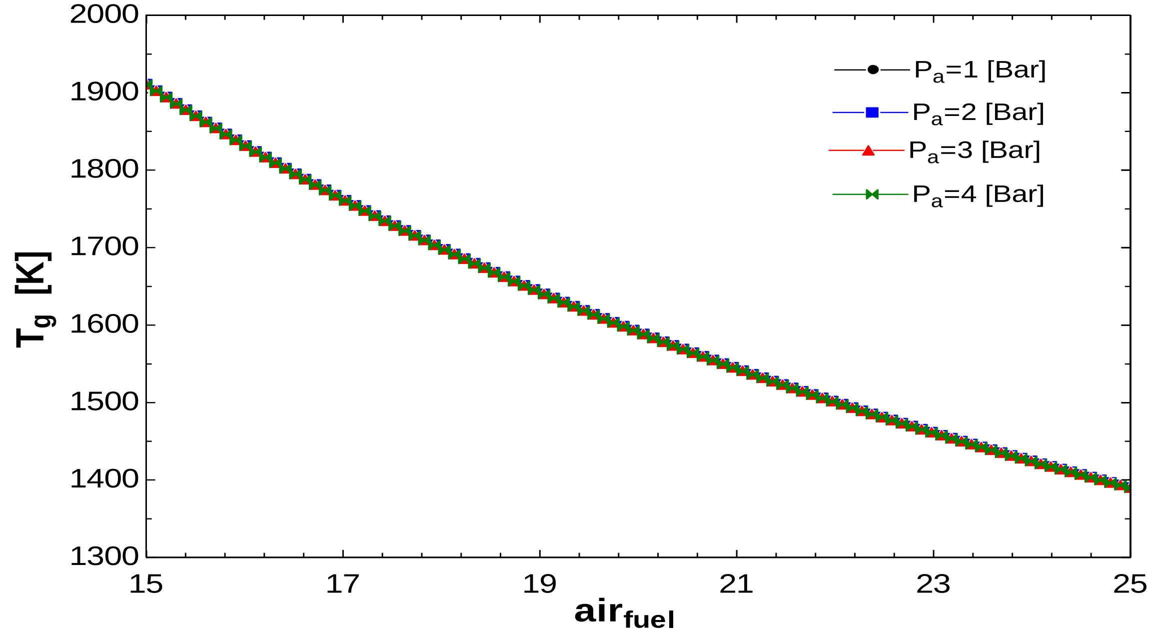

Fig. 6.1 Hot gas temperature in furnace VS A/F ratio (at different air pressure)

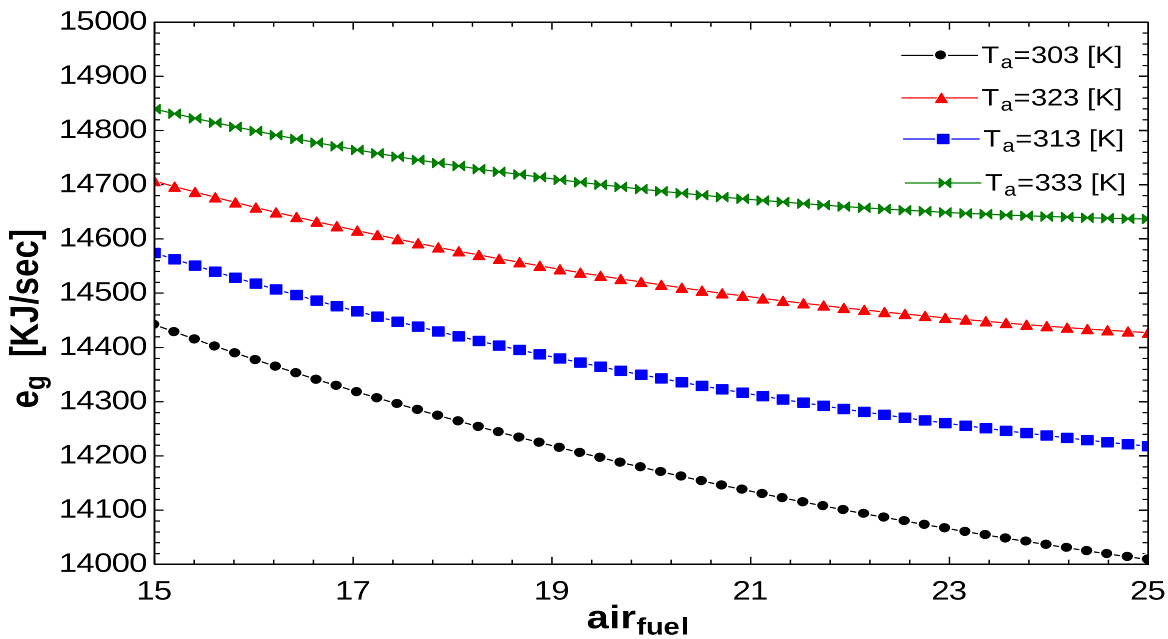

Fig. 6.2 Exergy of hot gas VS A/F ratio (at different air temperature.)

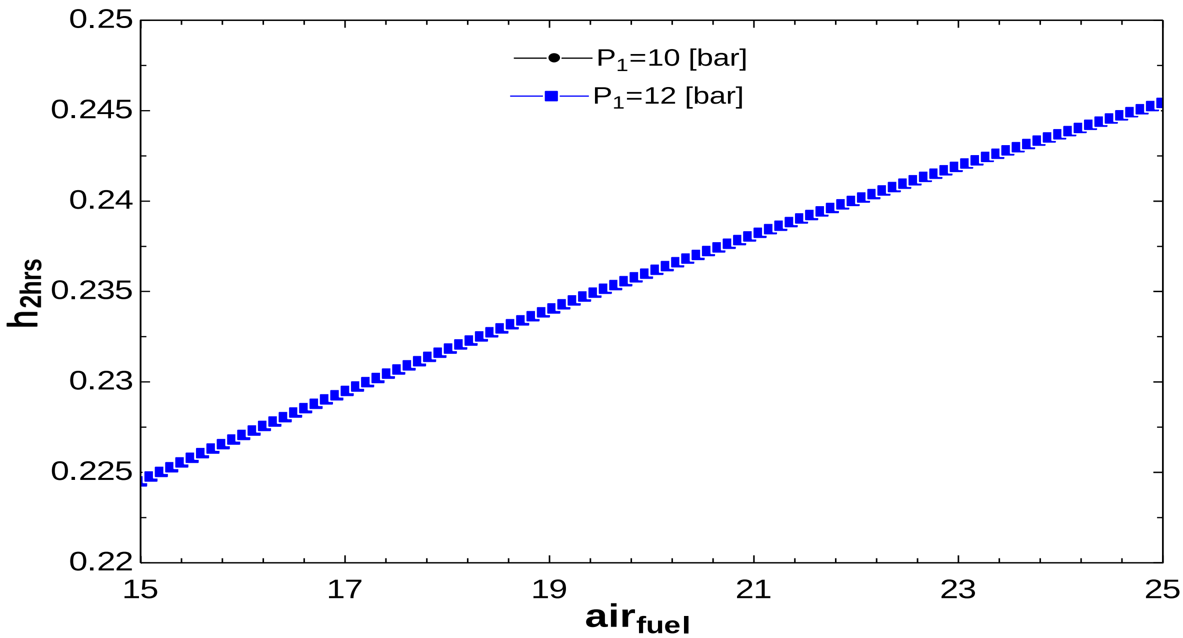

Fig. 6.3 Second law eff. of HRS VS A/F ratio (at different boiler Pressure)

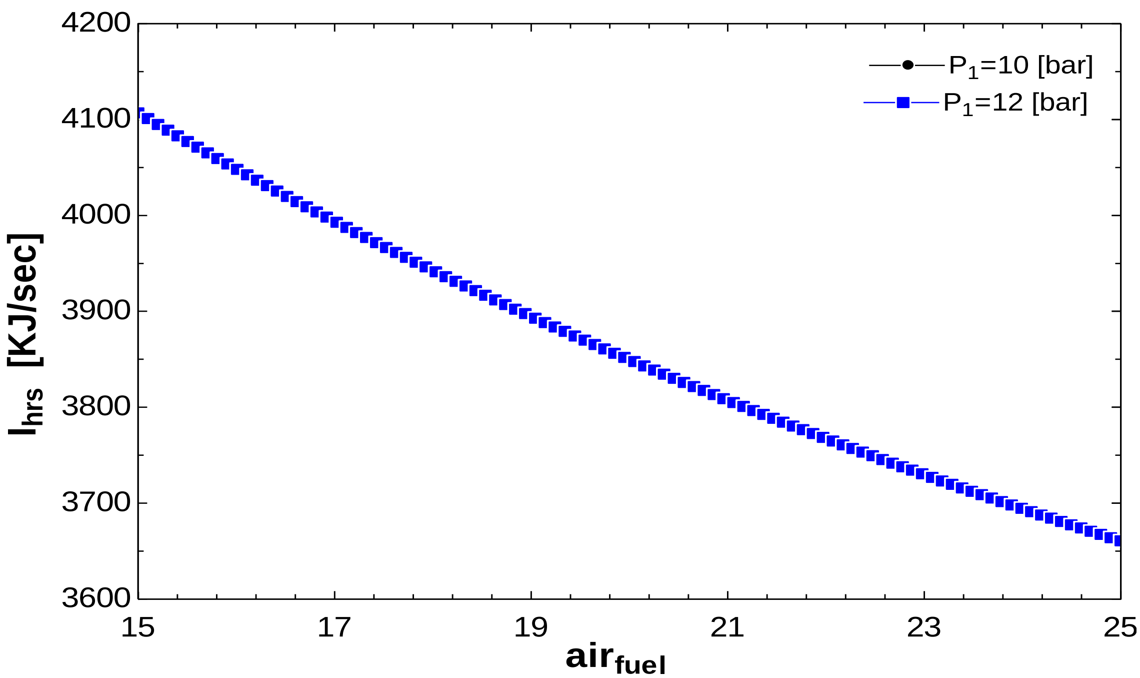

Fig. 6.4 Exergy destruction rate of HRS VS A/F ratio (at different boiler Pressure)

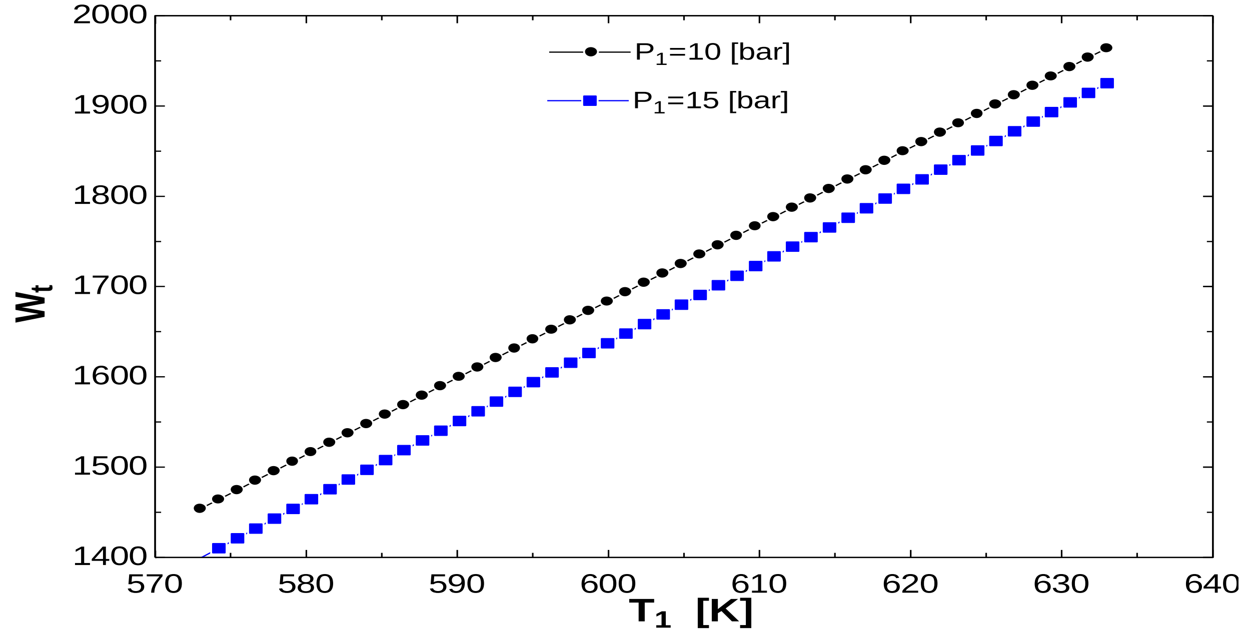

Fig. 6.5 Work output of turbine VS turbine inlet temperature T_1 (at different boiler Pressure).

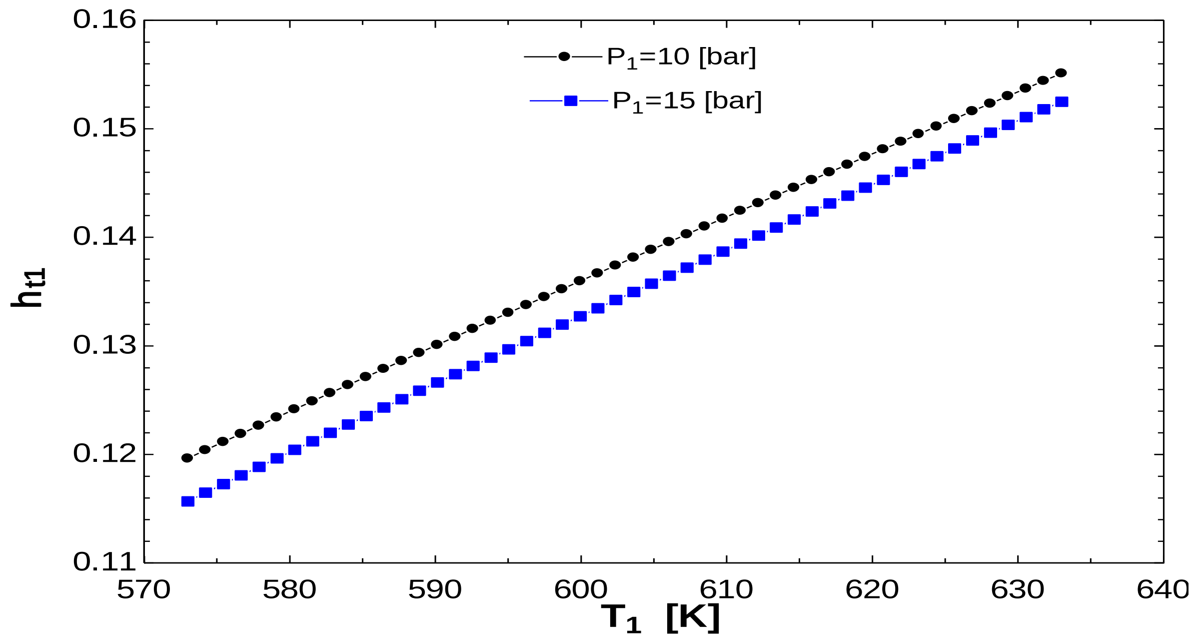

Fig. 6.6 First law eff. of turbine VS turbine inlet temperature T_1 (at different boiler Pressure)

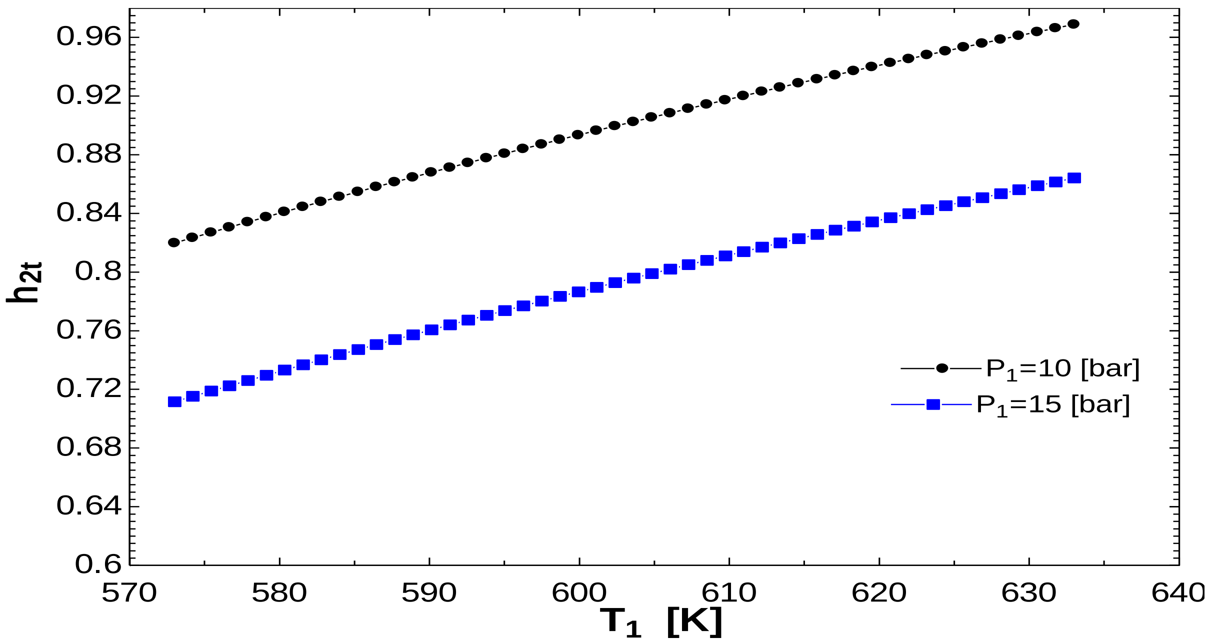

Fig. 6.7 Second law eff. of turbine VS turbine inlet temperature T_1 (at different boiler Pressure)

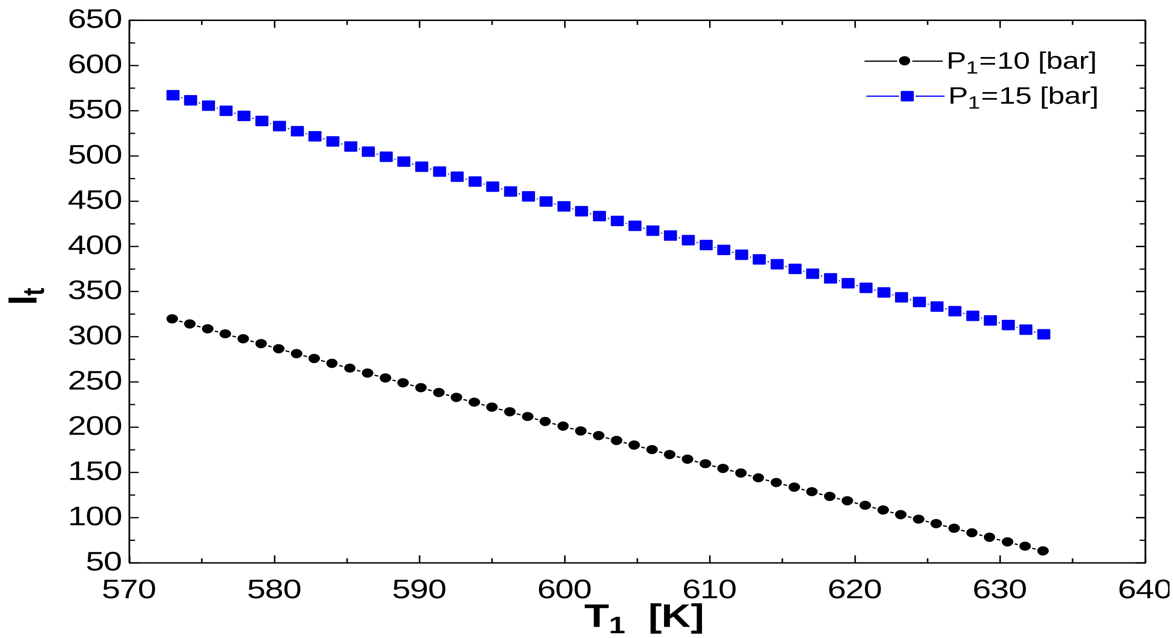

Fig. 6.8 Exergy destruction in turbine VS turbine inlet temperature T_1 (at different boiler Pressure)

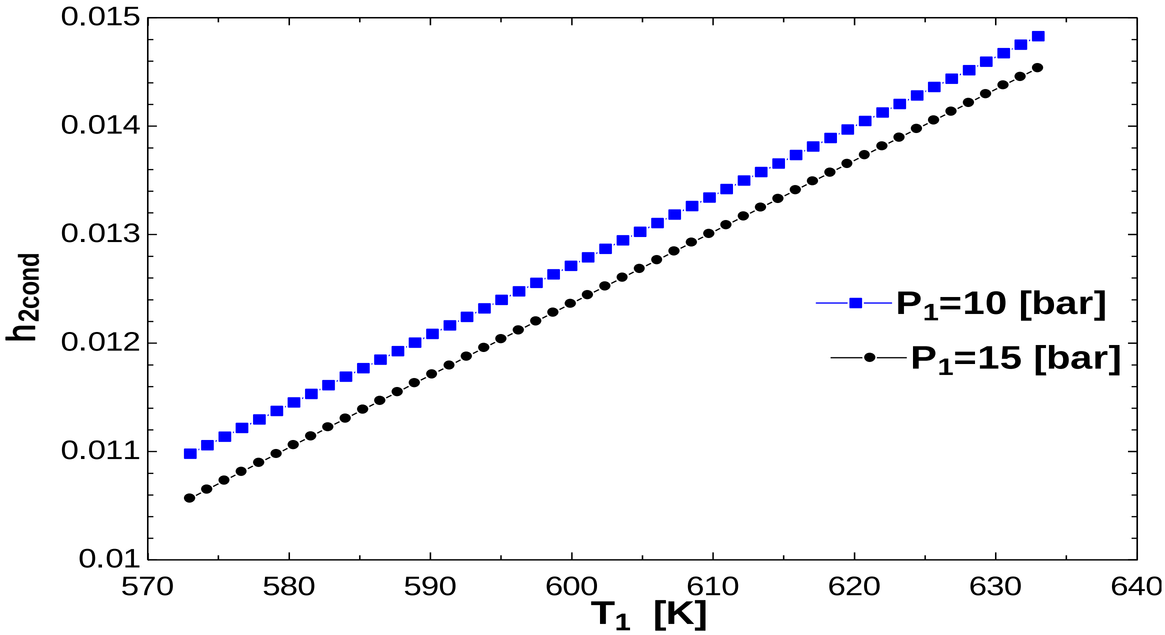

Fig. 6.9 Second law eff. of condenser VS T_1 (at different boiler Pressure)

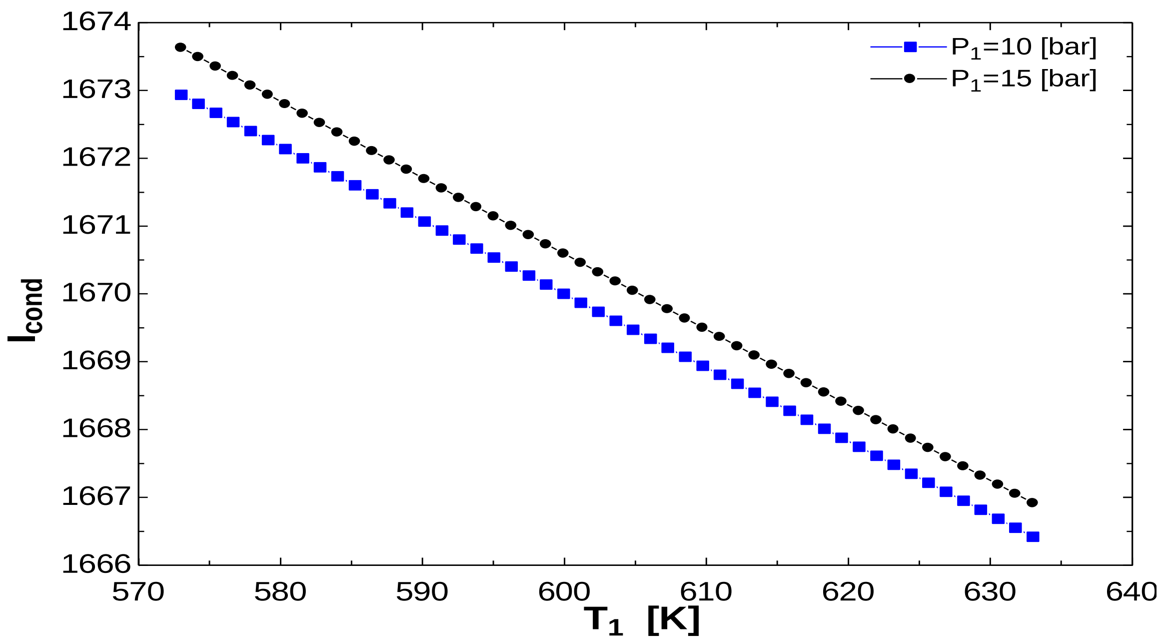

Fig. 6.10 Exergy destruction of condenser VS T_1 (at different boiler Pressure)

CHAPTER-7

CONCLUSION AND FUTURE WORK

CONCLUSION:

In this study, energetic and exergetic analyses of coal fired steam power plant have been performed. Thermodynamic analyses of different component of the steam power plant have been carried out. Exergy analysis is shown in this thesis to be able to help understand the performance of coal fired steam power plants. Variation of the first law efficiency, exergetic efficiency, and exergy destruction rate with respect to steam pressure, steam temperature and air fuel ratio have been analysis with the help of different figures. But in this present work, the transmission losses are not considered.

Following conclusion has been from this analysis of steam power plant:

- By the exergy analysis of different components of steam power plant, it is found that in condenser there is the maximum losses occur this is because of the phase conversion process of the working fluid. The exergy destruction measures indicate that the Condenser has the highest influenced component on the overall system, and thus, this more significant component is taken into consideration. Therefore, this thesis focuses on the parameters in optimization and suggests that a decrease in exergy destruction of condenser can improve the system performance.
- The second law efficiency has been found the least in case of condenser i.e. minimum exergetic efficiency. So the maximum amount of useful energy from condenser is being wasted. To, increase the overall performance of the system, it is very much necessary to minimize the wastage of useful energy (exergy) from this component followed by the HRS and steam turbine.

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APPENDICES

1. Thermodynamic Properties of steam

t, °C	1.0 MPa (t_s = 179.878 °C)			
	v	ρ	h	s
t_s(L)	1.127 23	887.13	762.52	2.1381
t_s(V)	194.36	5.1450	2777.1	6.5850
0	0.999 70	1000.30	0.98	-0.000 09
5	0.999 59	1000.41	22.01	0.076 24
10	0.999 87	1000.13	42.99	0.151 00
15	1.000 48	999.52	63.94	0.224 31
20	1.001 38	998.62	84.85	0.296 28
25	1.002 55	997.45	105.75	0.366 97
30	1.003 97	996.05	126.64	0.436 45
35	1.005 60	994.43	147.53	0.504 78
40	1.007 44	992.61	168.41	0.572 02
45	1.009 48	990.61	189.30	0.638 19
50	1.011 71	988.43	210.19	0.703 35
55	1.014 11	986.09	231.09	0.767 53
60	1.016 69	983.59	252.00	0.830 77
65	1.019 43	980.95	272.92	0.893 10
70	1.022 33	978.16	293.86	0.954 55
75	1.025 39	975.24	314.81	1.0152
80	1.028 60	972.19	335.77	1.0750
85	1.031 97	969.02	356.75	1.1340
90	1.035 50	965.72	377.76	1.1922
95	1.039 17	962.30	398.79	1.2497

100	1.043 00	958.77	419.84	1.3065
105	1.046 99	955.12	440.92	1.3626
110	1.051 12	951.36	462.04	1.4181
115	1.055 42	947.49	483.19	1.4729
120	1.059 87	943.51	504.38	1.5272
125	1.064 49	939.42	525.60	1.5808
130	1.069 27	935.21	546.88	1.6339
135	1.074 23	930.90	568.20	1.6865
140	1.079 35	926.48	589.58	1.7386
145	1.084 66	921.95	611.01	1.7901
150	1.090 15	917.31	632.50	1.8412
155	1.095 83	912.55	654.06	1.8919
160	1.101 71	907.68	675.70	1.9421
165	1.107 80	902.69	697.41	1.9919
170	1.114 10	897.58	719.20	2.0414
175	1.120 63	892.35	741.08	2.0905
180	194.44	5.1431	2777.4	6.5857
185	197.42	5.0653	2790.7	6.6148
190	200.34	4.9916	2803.5	6.6427
195	203.20	4.9212	2816.0	6.6695
200	206.02	4.8539	2828.3	6.6955
210	211.56	4.7268	2852.2	6.7456
220	216.98	4.6087	2875.5	6.7934
230	222.31	4.4983	2898.4	6.8393
240	227.56	4.3944	2920.9	6.8836
250	232.75	4.2965	2943.1	6.9265
260	237.88	4.2038	2965.1	6.9681

270	242.96	4.1159	2986.9	7.0087
280	248.01	4.0322	3008.6	7.0482
290	253.01	3.9524	3030.2	7.0868

	1.0 MPa (t_s = 179.878 °C)			
t, °C	v	ρ	h	s
300	257.99	3.8762	3051.6	7.1246
310	262.94	3.8032	3073.0	7.1616
320	267.86	3.7333	3094.4	7.1979
330	272.76	3.6662	3115.7	7.2335
340	277.64	3.6018	3136.9	7.2685
350	282.50	3.5398	3158.2	7.3029
360	287.35	3.4801	3179.4	7.3367
370	292.18	3.4225	3200.7	7.3700
380	297.00	3.3670	3221.9	7.4028
390	301.81	3.3133	3243.2	7.4351
400	306.61	3.2615	3264.5	7.4669
410	311.39	3.2114	3285.8	7.4984
420	316.17	3.1629	3307.1	7.5294
430	320.94	3.1159	3328.5	7.5600
440	325.69	3.0704	3349.9	7.5902
450	330.45	3.0262	3371.3	7.6200
460	335.19	2.9834	3392.8	7.6495
470	339.93	2.9418	3414.3	7.6786
480	344.66	2.9014	3435.8	7.7075
490	349.39	2.8621	3457.4	7.7360
500	354.11	2.8240	3479.1	7.7641
520	363.54	2.7507	3522.6	7.8196

540	372.95	2.6813	3566.2	7.8740
560	382.35	2.6154	3610.1	7.9273
580	391.74	2.5527	3654.2	7.9796
600	401.11	2.4931	3698.6	8.0310
620	410.47	2.4362	3743.2	8.0815
640	419.82	2.3820	3788.0	8.1312
660	429.16	2.3301	3833.1	8.1800
680	438.50	2.2805	3878.5	8.2281
700	447.83	2.2330	3924.1	8.2755
720	457.15	2.1875	3970.0	8.3221
740	466.47	2.1438	4016.1	8.3681
760	475.78	2.1018	4062.5	8.4135
780	485.08	2.0615	4109.2	8.4582
800	494.38	2.0227	4156.1	8.5024
820	503.68	1.9854	4203.3	8.5460
840	512.97	1.9494	4250.8	8.5890
860	522.26	1.9147	4298.5	8.6315
880	531.55	1.8813	4346.5	8.6735
900	540.83	1.8490	4394.8	8.7150
920	550.11	1.8178	4443.3	8.7560
940	559.39	1.7877	4492.1	8.7965
960	568.67	1.7585	4541.1	8.8366
980	577.94	1.7303	4590.4	8.8763
1000	587.21	1.7030	4639.9	8.9155
1100	633.54	1.5784	4891.4	9.1056
1200	679.83	1.4710	5148.9	9.2866
1300	726.10	1.3772	5411.9	9.4593

1400	772.34	1.2948	5680.0	9.6245
1500	818.57	1.2216	5953.0	9.7830
1600	864.78	1.1564	6230.3	9.9351
1800	957.19	1.0447	6796.7	10.222
2000	1049.6	0.952 78	7376.8	10.489