EXERGO-ECONOMIC ANALYSIS OF THERMAL POWER PLANT

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

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IN

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CHANDAN GUPTA

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Under the Guidance of

Dr. B. B. ARORA

Associate Professor

DEPARTMENT OF MECHANICAL, PRODUCTION & INDUSTRIAL AND AUTOMOBILE ENGINEERING DELHI TECHNOLOGICAL UNIVERSITY, DELHI-110042

DECLARATION

I hereby declare that the work which being presented in the major thesis entitled **"EXERGO-ECONOMIC ANALYSIS OF THERMAL POWER PLANT"** in the partial fulfilment 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'. I have not committed plagiarism while completing the attached piece of work, similarity found after checking is ___ which is below the permitted limit of 20%.

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CERTIFICATE

This is to certify that CHANDAN GUPTA, (Roll no. 2K14/THE/06), student of M.Tech, THERMAL ENGINEERING, Delhi Technological University, has submitted the dissertation titled **"Exergo-economic analysis of Thermal power plant"** under our guidance towards the partial fulfilment of the requirements for the award of the degree of Master of Technology under our guidance and supervision.

DR. B. B. ARORA
Associate Professor
Department of Mechanical Engineering
Delhi Technological University
(Formerly Delhi College of Engineering)
Delhi-110042

ACKNOWLEDGMENT

It is said that gratitude is a virtue. This part is dedicated to special thanks that I would like to deliver to the people who helped me in making the fulfilment of this thesis project possible.

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CHANDAN GUPTA
(2K14/THE/06)

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 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. The primary objectives 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 parameters such as pressure, temperature and mass flow rate. The effects of theses parameters on the system performances are investigated.

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LIST OF SYMBOLS USED

$\dot{m}$	mass flow rate (kg/sec)
h	specific enthalpy (kJ/kg)
To	surrounding temperature (Kelvin)
s	specific entropy (kJ/kg K)
I	exergy destruction (kJ/sec)
X	exergy rate (kJ/sec)
W	power (KW)
e	specific exergy (KJ/Kg)
x	mole fraction of constituents
Ci	Component cost (INR)
Cdirect	Direct plant cost (INR)
CEqu	Cost of equipment (INR)
COther	Other cost due to direct installation, instrumentation, auxiliary services, and control & site preparation (INR)
Cpiping	Cost of piping (INR)
Cashhandling	Cost of ash handling (INR)
Ccoalhandling	Cost of coal handling (INR)
$g_{conversion}$	A conversion factor

g_{MW}	Percentage of net electric energy power plant output assumed as 90 % of MW available for sale.
μ	Factor for other cost
Y	Factor that accounts for engineering and plant startup
Cindirect	Indirect plant cost (INR)
Ctci	Total capital investment cost (INR)
i	Interest rate
j	jth year
Ccoal	Coal cost (INR)
m_{coal}	Mass consumption rate of coal at the jth year (tonne/h)
Cc	Fuel cost per tonne (INR)
Clab	Cost of labour (INR)
NL	No. of labour employed
Cs	Labour average annual salary(INR/year)
Cmaint	Maintenance cost (INR)
Cins	Insurance cost (INR)
Cpumping	Pumping cost
n	Plant life in year
N	Number of pump
Cep	Cost of electricity
CO	Total operating cost
Ran	Revenue earned (INR)

LIST OF SUFFIX

f	fuel
pa	primary air
sa	secondary air
g	hot gas stream
i	input
o	output
sup	superheated steam
t	Turbine
p	pump
con	condenser
cc	combustion chamber
cw	cold water through condensor
com	compressor
t	turbine
ph	physical
ch	chemical

GREEK SYMBOL

η_I	first law efficiency
η_{II}	second law efficiency
ε	exergy associated with individual's component (kJ/sec)
X	exergy rate (kJ/sec)

CHAPTER-1

INTRODUCTION

1. FUNCTIONAL DISCRIPTION:

In Thermal Power Station fuel burns and use the resultant to make the steam, which derives the turbo generator. The Fuel (solid or fluid) is burnt in the furnace. The pressure energy of steam produce is converted into mechanical energy with the help of turbine. The mechanical energy is given to the generator where the magnet rotate inside a stator winding & thus electricity is produced. In India 65% of total power is generated by thermal power stations. To understand the working of the Thermal Power Station plant, we can divide the whole system into following parts**.**

1.1 Coal flow:

In coal fired plants, raw material are air & water, coal is transported through Railway wagons from Coal companies & is kept reserved on a buffer stock. This brought out to the station is unloaded with the help of wagon tippler. After unloading, the coal is sent to crusher house with the help of conveyor belts. The coal which is now reduced to very small pieces is sent to the coal bunkers with the help of conveyor belt. The raw coal is fed to coal mills through raw coal feeders raw coal feeders basically regulate raw coal to pulverized coal pipes. A position of the primary air is heated utilizing the heat of the fuel gases & then mixed with the cold air as per requirement by the pulverized coal. Normally the temperature is maintained at 60 to 70 degrees. The coal is now burnt in the furnace using oil in the beginning showered through the nozzles at different elevations in the furnace. To provide air for combustion, the heat of the flue gases also heat it the heat produced due to combustion is utilized for the conversion of water into steam. This water is stored in the boiler drum. There are two sets of pipes attached to the drum, one called riser & other known as down corner through which the water comes to the ring header & steam moves up due to the density difference of water & steam. Its steam is super heated using super heaters & meanwhile the flue gases are throughout in the atmosphere through chimney.

1.2. Steam flow:

The super heated steam is sent to the turbine through pipelines there are three turbines in the units, using this steam at different temperature & pressures. After passing through high pressure turbine the steam is send to the reheater for rising the temperature of the steam. After reheating the steam is sent to the intermediate pressure turbine through reheated line. Here it losses most of its temperature & pressure & finally sent to low pressure turbine. The uses of three different turbines help in increasing the efficiency of the plant. The turbine in turn connecting with a generator produces electricity. Then this electricity is stepped up to 220 KV with the help of step up transformer & supplied to various sub-stations grids.

Meanwhile, the steam through low pressure (L.P.) Turbine is condensed and the condensed water is stored in hot well.

1.3. Water flow:

The condensed water is extracted from the hot well through condensate extraction pumps & sent to the boiler drum with the help of BOILER FEED PUMP (B.F.P.) before passing through low pressure heater and dearater. While loss in water is make up from C.S. Tank, which have D.M. Moor in it. The C.S. Tank is directly connected to hot well. The water used in condenser is sent to cooling tower for cooling. After cooling this water is again sent to condenser with the help of circulating water pump. The loss is making from raw water pump house through clarifier pump house.

2. COMPONENT DISCRIPTION:

2.1. WAGON TIPPLER: It is the machine which is used to tip the coal from the wagon. The coal tipped is directly feed to conveyor belt. Its capacity is 12 wagon per hour.

2.2. CRUSHER: It crushes the coal into small pieces.

2.3. COAL MILLS: In it small pieces of coal are converted into pulverized from. They are 6 in number.

2.4. FURNACE: It is the chamber in which fuel burns & fire blows.

2.5. BOILER DRUM: It contains water for boiling.

2.6. ELECTROSTATIC PRECIPITATOR: In this we have electrodes which attract fly ash and extract it from flue gases so that it cannot enter atmosphere.

2.7. CHIMENY: It is used to release flue gases into the atmosphere.

2.8. TURBINE: Turbine is the part which revolves due to steam pressure. It is of three types.

a) High pressure turbine.

b) Intermediate pressure turbine.

c) Low pressure turbine.

2.9. TURBO GENERATOR: It is the main machine which produces 250 MW electricity .It is (H_2O) water and H_2 (Hydrogen) gas cooled therefore it is contained in cylindrical chamber.

2.10. CONDENSER: It condenses steam coming from low pressure turbine (L.P.T.) to hot water. By removing air and other non-condensable gases from steam while passing through them.

2.11. COOLING WATER (C.W.) PUMP: This pump send water from cooling tower to condenser.

2.12. COOLING TOWER: It is used to coal the water its height is near about 143.5 meter. The hot water is led to the tower top and falls down through the tower and is broken into small particles while passing over the baffling devices. Air enters the tower from the bottom and flow upwards. The air vaporizes a small percentage of water, thereby cooling water falls down into tank below the tower from where it is pumped to the condenser and cycle is repeated.

2.13. RAW WATER PUMP HOUSE: It supplies raw water to the boiler**.**

2.14. CLARIFIER PUMP HOUSE: The water from raw is clear at clarifier by putting alum in it & filtering it & then supplied to the condenser.

2.15. CONDENSATE EXTRACTION PUMP: C.E.P. pump is used to extract the condense water from the hot well and supply to the deaerator after passing through L.P. heater & Economizer, so that high pressure steam in the cylinder can be created.

2.16. LOW PRESSURE HEATER: It is used to increase the temperature of water, in this way efficiency of system increases.

2.17. DEAREATER: It is used to remove air from water which is entrapped in the water molecules. It is very important part because the entrapped air effect air drum badly.

2.18. BOILER FEED PUM (B.F.P.): It is the heaviest drive in the plant & supply water to boiler drum from dearator.

2.19. HIGH PRESSURE HEATER (H.P.): In this temperature of water increases. Thus efficiency further increases.

2.20. ECONOMISER: In this flue gases exchange heat to the water to increase system efficiency, causes saving in fuel consumption (5 to 10%). Economizer tubes are made up of steel either smooth or covered with fins to increase the heat transfer surface area.

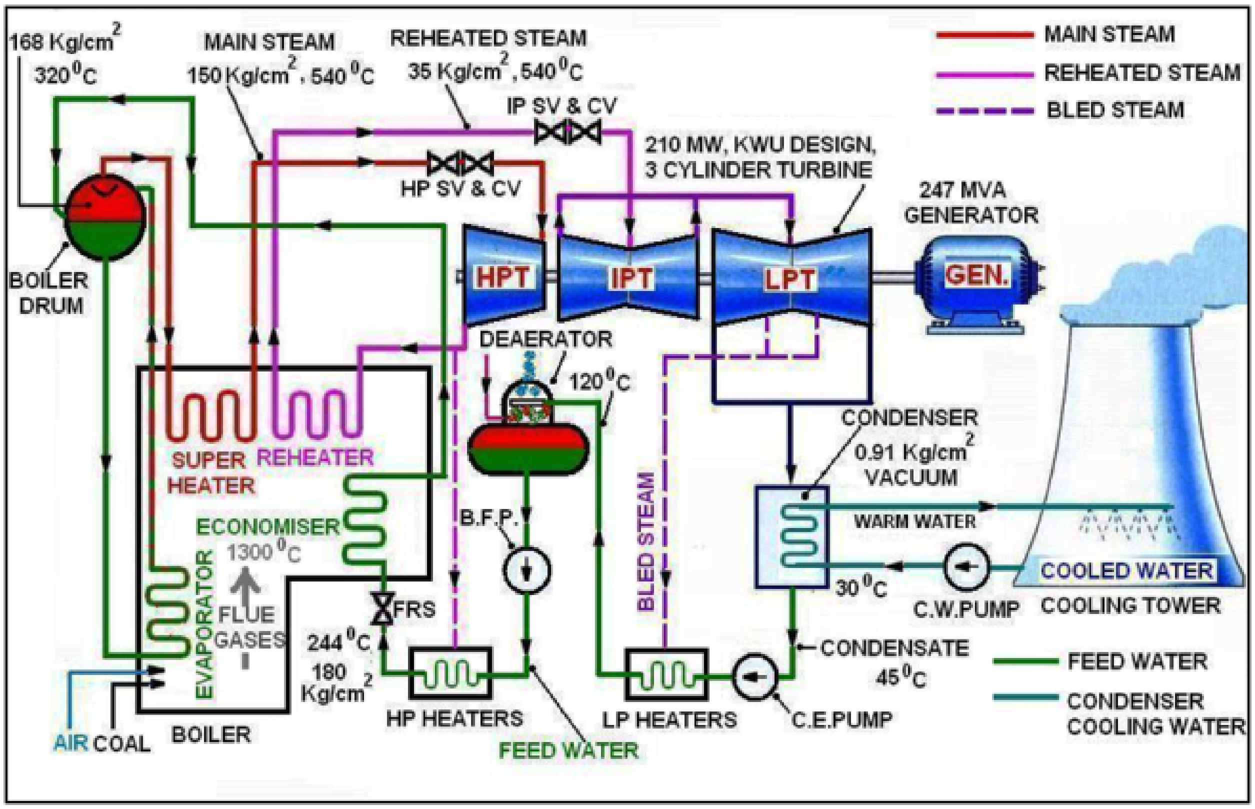
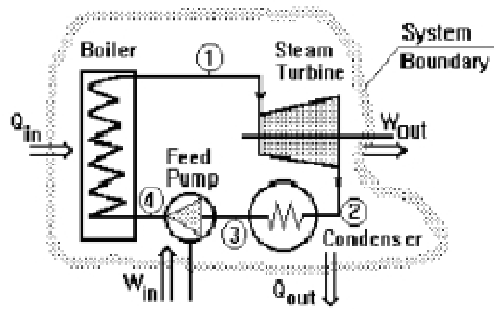
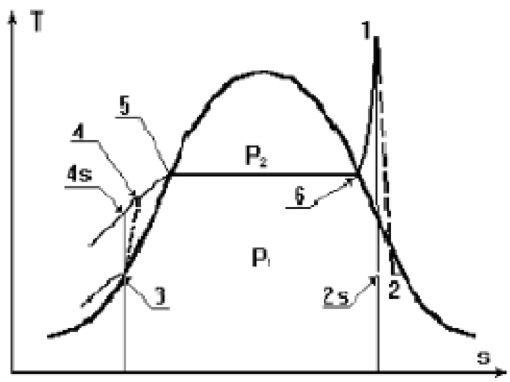


Figure.1.1 Component of Thermal power plant.

3. WORKING CYCLE: Rankine Cycle- The Rankine cycle is a heat engine with a vapour power cycle. The common working fluid is water.





Figures 1.2 (a) and 1.2 (b) Simple Rankine cycle

1 to 2: Isentropic expansion (Steam turbine), an isentropic process, in which the entropy of working fluid remains constant.

2 to 3: Isobaric heat rejection (Condenser) An isobaric remains constant.

3 to 4: Isentropic compression (Pump) during the isentropic compression process, external work is done on the working fluid by means of pumping operation.

4 to 1: Isobaric heat supply (Steam Generator or Boiler) during this process, the heat from the high temperature source is added to the working fluid to convert it into superheated steam.

According to the T-s diagram shown in

Figure 1.2(b), the work output W1 during isentropic expansion of steam in the turbine, and the work input W2 during isentropic compression of working fluid in the pump are:

W1 = m (h1 – h2)

W2 = m (h4 – h3)

Where m is the mass flow of the cycle and h1, h2, h3, h4 is enthalpy. Heat supplied to the cycle (steam generator or boiler) Q1, and heat rejected from the cycle (condenser) Q2, are:

Q1 = m (h1-h4)

Q2 = m (h2-h3)

The net work output of the cycle is:

W = W1 – W2

The efficiency of the Rankine cycle is:

η = W/Q1

as high as a Carnot cycle but the cycle has less practical difficulties and is found to be more economical. Thermodynamically, it is true that no heat engine can be more efficient than a reversible heat engine working between two fixed temperature limits.

A Carnot cycle consists of the following processes as shown in Figure 2. These processes are theoretical because these cannot be achieved practically due to certain limitations. A brief discussion of these processes is discussed below. Theoretically, the

Carnot cycle is the most efficient heat energy conversion cycle.

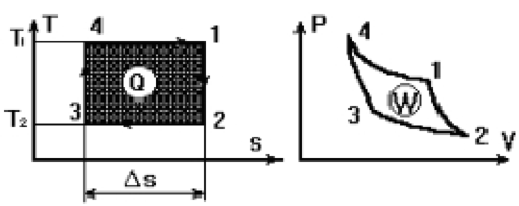


Figure 1.3: T-s and P-V diagrams for Carnot cycle.

1 to 2: Isentropic expansion,

2 to 3: Isothermal heat rejection,

3 to 4: Isentropic compression,

4 to 1: Isothermal heat supply

The heat supplied to the cycle per unit mass flow,

$Q1 = T1\ ds$,

And the heat rejected from the cycle per unit mass flow,

$Q2 = T2\ ds$.

By applying the first law of thermodynamics to the cycle, we obtain:

$Q1 – Q2 – W = 0$

And the thermal efficiency of the cycle will be:

$\eta = W/Q1 = 1 – T2 / T1$

Due to mechanical friction and other irreversibility's, no cycle can achieve this efficiency. The gross work output of the cycle, i.e. the work done by the system is:

$Wg = W4\text{-}1 + W1\text{-}2$ (1.9)

And the work ratio is defined as the ratio of the net work W to the gross work output Wg, i.e.

W/Wg.

The Carnot cycle has a low work ratio.

Although, theoretically, this cycle is the most efficient. System for power generation, it cannot be used in practice. There are several reasons such as a low work ratio, economical aspects and practical difficulties. Some recent developments to enhance the efficiency of the Rankine cycle have been analyzed.

Thermal power plant based on a Rankine cycle: In a simple Rankine cycle, steam is used as the working fluid, generated from saturated liquid water (feed-water). This saturated steam flows through the turbine, where its internal energy is converted into mechanical work to run an electricity generating system. All the energy from steam cannot be utilized for running the generating system because of losses due to friction, viscosity, bend-on-blade etc. Most of the heat energy is rejected in the steam condenser. The feed water brings the condensed water back to the boiler. The heat rejected during condensation of steam in the condenser is given away by a sink. As a result of the conversion of much of its thermal energy into mechanical energy, or work, steam leaves the turbine at a pressure and temperature well below the turbine entrance values. Basically the low- pressure steam leaving the turbine at state 2 is first condensed to a liquid at state 3 and then pressurized in a pump to state 4, and this high pressure liquid water is then ready for its next pass through the steam generator to state 1 and is reused around the Rankine cycle again as shown in Figure 3. The steam generator and condenser both function as heat exchangers. In a well-designed heat exchanger, both hot and cold fluids flow with little pressure loss. Therefore, ideally it can be considered that steam generators and condensers have negligible pressure loss, where fluids are operating without any change in pressure. Hence, the Rankine cycle is operating between two fixed pressure levels, i.e. the pressure in the steam generator and pressure in the condenser. A pump provides the pressure increase, and a turbine provides the controlled pressure drop between these levels. The Rankine cycle as a system converts thermal energy of the steam into mechanical energy by means of a turbine rotation, which runs the generator to produce electrical energy.

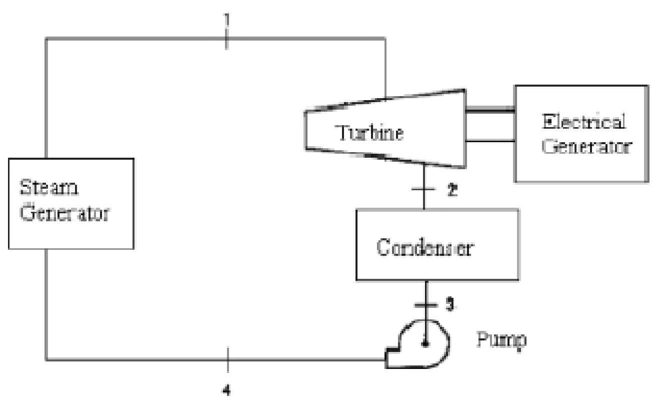


Figure 1.4: Rankine-cycle and T-s diagrams.

3.1. Rankine cycle analysis

In the analysis of a heat engine cycle, it is assumed that the fluid flow follows the continuity equation applicable to steady state conditions. All the power plants are assumed to run under steady state conditions. The starting and shutting down working condition is exempted where deviations from a steady state cannot be avoided. With reference to the Rankine cycle shown in Figure 3, the control of the flow can be exercised by a valve.

Partial closer of the valve would reduce the flow of steam in the turbine and power output. In the ideal and the steam generator heat transfer is given by the equation;

Qa = h1 – h4 [kJ/kg]

The condenser is a large shell-and-tube type heat exchanger. This is positioned next to the turbine in order to receive a large flow rate of low pressure steam. This steam in the condenser goes under a phase change from vapour to liquid water.

External cooling water is pumped through thousands of tubes in the condenser to transport the heat of condensation of the steam away from the plant. Upon leaving the condenser, the condensate is at a low temperature and pressure. Removal of this condensate may be considered as maintaining the low pressure in the condenser continuously. The phase change in turn depends on the transfer of heat to the external cooling water. The rejection of heat to the surroundings by the cooling water is essential to maintain the low pressure in the condenser.

Applying the steady-flow First Law of Thermodynamics to the condensing steam enables:

$Qc = h3 – h2$ [kJ/kg]

The value of Qc is negative because $h2 > h3$. Thus, consistent with sign convention, qc represents an outflow of heat from the condensing steam. This heat is absorbed by the cooling water passing through the condenser tubes. The rise in cooling water temperature and mass-flow rate mc are related to the rejected heat by the following equation,

$ms|qc| = mc\ cwater\ (Tout – Tin)$ [kW]

Where cwater is the heat capacity of the cooling water [kJ/kg-K]. The condenser cooling water may be drawn from a river or a lake at a lower temperature and returned downstream at a higher temperature.

Alternatively, it may be circulated through cooling towers where heat is rejected from the cooling water to the atmosphere. The condenser heat transfer can be expressed in terms of an overall heat transfer coefficient, U, the mean cooling water temperature,

$Tm = (Tout + Tin) / 2$,

And the condensing temperature T3 by the following equation;

$ms|qc| = UA\ (T3 – Tm)$ [kJ/s]

For a given heat rejection rate, the condenser size is represented by the tube surface area A that depends inversely on (a) the temperature difference between the condensing steam and the cooling water, and (b) the overall heat-transfer coefficient U. For a fixed average temperature difference between the two fluids on opposite sides of the condenser tube walls, the temperature of the available cooling water controls the condensing temperature Rankine cycle, working fluid follows reversible an adiabatic or isentropic path in the turbine and is subjected to lower pressure and temperature in the condenser.

Applying the First Law of Thermodynamics for an isentropic turbine:

q = 0 = h2 – h1 + wt [kJ/kg]

Where kinetic and potential energy differences between the inlet and outlet are negligibly small.

The above equation shows that the turbine work per unit mass passing through the turbine is the difference between the entrance enthalpy and exit enthalpy,

wt = h1 – h2 [kJ/kg]

The power delivered by the turbine to an external load, such as an electrical generator, is given as,

Turbine power = ms wt = ms (h1 – h2) [kW]

Applying the steady-flow First Law of Thermodynamics to the steam generator, shaft work is zero and hence, the pressure of the condensing steam.

Therefore, the colder the cooling water, the lower is the temperature and pressure of the cycle and the higher is the thermal efficiency of the cycle. A pump is a device that moves liquid from a low pressure to high pressure. In the Rankine cycle, the condensate is raised to the pressure of the steam generator by boiler-feed-pumps (BFP). The high-pressure liquid water entering the steam generator is called feed water.

From the steady-flow First Law of

Thermodynamics, the work and power required to drive the pump are given by the equations,

Work, wp = h3 – h4 [kJ/kg]

And,

Pump power = mswp = ms (h3 – h4) [kW]

The pump work has a negative value as h4 > h3.

This is consistent with the thermodynamic sign convention, which indicates that work and power must be supplied to operate the pump. The net power delivered by the Rankine cycle is the difference between the turbine power and the magnitude of the pump power.

One of the significant advantages of the Rankine cycle is that the pump power is usually quite small compared with the turbine power. This is indicated by the work ratio, wt / wp, which is large compared with one for the Rankine cycle. As a result, the pumping power is neglected in approximating the Rankine cycle net power output. It is assumed that the liquid at a pump entrance is saturated liquid because the condensate temperature never reaches below exit cooling water temperature. That is usually the case for power-plant feed-water pumps.

Sub cooling would increase the heat input in the steam generator, and on the other hand, the introduction of steam into the pump would cause poor performance. The properties of the pump inlet or condenser exit therefore, may be obtained directly from the saturated-liquid curve at the (usually) known condenser pressure. The properties for an isentropic pump discharge at state 4 could be obtained from a sub cooled-water property table at the known inlet entropy and the throttle pressure. However, such tables are not widely available. The enthalpy of a sub cooled state is commonly approximated by the enthalpy of the saturated- liquid evaluated at the temperature of the sub cooled liquid. This is usually quite accurate because the enthalpy of a liquid is almost independent of pressure as the Rankine cycle based thermal power plant works at atmospheric pressure.

A measure of the effectiveness of an energy conversion device is its thermal efficiency. This is defined as the ratio of the cycle network to the heat supplied from external sources. Thus, by using the above equations, the ideal Rankine-cycle thermal efficiency in terms of cycle enthalpies is given as:

$\eta = (h1 - h2 + h3 - h4)/ (h1 - h4)$

In accordance with the Second Law of Thermodynamics, the Rankine cycle efficiency must be less than the efficiency of a Carnot engine operating between the same temperature extremes. As with the Carnot-cycle efficiency, Rankine-cycle efficiency improves when the average heat-addition temperature increases and the heat rejection temperature decreases. The cycle efficiency may be improved by increasing turbine inlet temperature and decreasing the condenser pressure (and thus the condenser temperature).

Another practical aspect affecting the efficiency is the heat rate. Heat rate is the ratio of the rate of heat addition in conventional heat units (boiler) to the net power output in conventional power units (turbine). Because the rate of heat addition is proportional to the fuel consumption rate, the heat rate is a measure of fuel utilization rate per unit of power output. In the United States, the rate of heat addition is usually stated in Btu/hr, and electrical power output in kilowatts, resulting in heat rates being expressed in kW-hr. It may be verified that the heat rate in English units is given by the conversion factor, 3413 kW-hr, divided by the cycle thermal efficiency as a decimal fraction, and that its value has a magnitude of the order of 10 000 kW-hr. In SI system of units, the heat rate is usually expressed in kJ/kW-hr, is given by 3600 divided by the cycle efficiency as a decimal fraction, and is of the same order of magnitude as in the English system. The low value of heat rate represents high thermal efficiency and is of course desirable.

4. Advantages and disadvantages of Thermal power plant:

Thermal power has been used as a natural form of energy for thousands of years in cooking and heating. Hot springs are just one example of this naturally-occurring thermal energy. With today's rising power demands, geothermal power plants are attractive options for their cheap, environmentally-friendly energy production. However like all power sources, thermal isn't perfect, and disadvantages temper the strengths.

4.1. Financial Costs:

One of the primary advantages of thermal power is that the generation costs are extremely low. No fuel is needed to generate the power, and the minimal energy needed to pump water to the Earth's surface can be taken from the total energy yield. Even considering transport, geothermal energy is estimated to save 80 percent of the costs associated with fossil fuels, such as oil and natural gas. The principal financial disadvantage of a geothermal system is its high initial installation costs. The longer a plant is operational, the more it pays for itself in the long run.

4.2 Environmental Impact:

Thermal power is revered by environmental activists because it is completely renewable, does not use fuel to produce power and has virtually no emissions. It also helps reduce global warming and pollution and requires far less land than a coal mine or oil field. The only environmental disadvantage is the occasional release of harmful gases. Since thermal power operates by drilling into the Earth's mantle, some poisonous gases can escape. These gases can be a danger to plant workers, who must wear protective equipment, but have little impact once dispersed into the atmosphere.

4.3 Job Creation and Hazards:

Thermal power facilities create a number of jobs for local communities. Researchers, scientists and drilling workers are among the specialists needed for safe and effective operations. Disadvantages in this area involve limited workplace hazards, such as crystalline silica dust and exposure to extremely hot steam and water mains. Fortunately, these dangers are minimal, especially compared to other energy industries, like fossil fuels.

4.4 Location:

A main disadvantage of thermal power plants is that they can only be constructed in areas where temperatures below the Earth's surface allow for the production of steam over a long period of time. The type of rock in the region must also be easy to drill through. Extensive research is needed to find these key areas; and because of their rarity, plants are sometimes forced to operate in relatively remote regions. Both of these factors contribute to the high initial cost of starting up a geothermal facility.

4.5 Long-term Viability and Risks:

In areas where there is long-term steam production, thermal plants can thrive and produce many megawatts of clean, renewable power. In these scenarios, plants quickly recoup initial costs. However, environmental factors can sometimes lead to a reduction in steam in a region, forcing companies to take the risk of potentially losing their investment. Lack of equipment or qualified personnel and the relative financial security of fossil fuel extraction by comparison serve as an added disadvantage, hampering the continued expansion of thermal power.

CHAPTER-2

LITERATURE REVIEW

2.1 Literature:

Sayed A. Abdel-Moneim et al. [1] shows the energy and exergy analysis of a combined cycle with a supercharged boiler was carried out. A combination of a basic gas turbine and steam cycle with both a supercharged boiler (SB) and a heat recovery boiler (HRB) was investigated. The effects of the inlet temperature of the gas turbine, the excess air factor, and the compressor pressure ratio on the performance of the supercharged boiler combined cycle (SBCC) were studied. Comparisons between the SBCC and the conventional combined cycle were performed. The results indicated that the SBCC gives output power up to 2.1 times of that of the conventional combined cycle when compared at the same values of the operating parameters. However, the SBCC efficiency was found to be lower than the conventional combined cycle. The exergy analysis showed an advantage of SBCC over the conventional combined cycle.

Mali Sanjay D et al [2] 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. Rashad et al. [3] In this study, the energy and exergy analysis of Shobra El-Khima power plant in Cairo, Egypt is presented. The primary objectives of this paper are to analyze the system components separately and to identify and quantify the sites having largest energy and exergy losses at different load. 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). Energy losses mainly occurred in the condenser where (404.653 MW at Max load, 306.747 MW at 75% load and 278.849 MW at 50% load) is lost to the environment. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the Turbine system (42% at Max load, 59% at 75% load and 46.1 at 50% load) followed by the condenser (28% at Max load, 20.3% at 75% load) while at 50% load the feed water heaters showed more exergy destruction (27.7%) than condenser (23.8) and then the feed water heaters (20.8% at max load, 12.1% at 75% load) . In addition, the calculated thermal efficiency based on the specific heat input to the steam was 43% while the exergy efficiency of the power cycle was (44% - 48%).

A. Valero et al. [4] resume the theoretical background supporting the main ideas of the exergy cost accounting and the thermoeconomic approach followed by Valero and co-workers. Part I introduces the basic requirements, with a simple example accompanying the dissertations, to calculate the exergy and thermoeconomic costs and to perform the thermoeconomic analysis of a complex system. The connections with other thermoeconomic approaches and schools are briefly explained. Part II presents, as an illustration of the applications of thermoeconomic analysis, some of the most interesting applications of costs to the operation diagnosis and optimization of a complex system.

Fathia Hafdhi et al. [5] An energetic and exergetic analysis is conducted on a Steam Turbine Power Plant of an existing Phosphoric Acid Factory. The heat recovery systems used in the different parts of the plant are also considered in the study. Mass, energy and exergy balances are established on the main compounds of the plant. 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 parameters such as pressure, temperature and mass flow rate. The effects of theses parameters on the system performances are investigated.

The main sources of irreversibility are the melters, followed by the heat exchangers, the steam turbine generator and the pumps. The maximum energy efficiency is obtained for the blower followed by the heat exchangers, the deaerator and the steam turbine generator. The exergy efficiency obtained for the heat exchanger, the steam turbine generator, the deaerator and the blower are 88%, 74%, 72% and 66% respectively. The effects of High Pressure steam temperature and pressure on the steam turbine generator energy and exergy efficiencies are investigated.

Amin M. Elsafi [6] shows Solar direct steam generation is considered as a promising technology for steam production in thermal power generation due to high temperature levels that can be achieved compared to other technologies that use indirect steam generation. This paper demonstrates exergy and exergoeconomic analysis of commercial-size direct steam generation parabolic trough solar thermal power plant. For steam power cycles, reheating might be necessary to avoid great wetness of steam which shortens the lifetime of the turbines. Therefore, two configurations have been considered in this study; the non-reheating configuration as well as reheating by steam–steam heat exchanger. For each component, exergy and exergy-costing balance equations have been formulated based on a proper definition of fuel–product–l loss. Exergy results show that particular attention should be paid to solar field, condenser, low pressure turbine and high pressure turbine (in a descendant order) as they constitute the major sources of exergy destruction. Results from exergoeconomic analysis, however, show that the condenser should be the fourth component in the order of importance after the solar field and low/high pressure turbines.

Increasing the temperature at the inlet of the low pressure turbine by 100 K using steam–steam reheating is shown to result in 9.1% increase in the vapor fraction at the exit of turbine. This increase in steam quality, however, would be achieved by drop less than 1.5% in thermal and exergetic efficiencies, and about 2% increase in cost of electricity. Moreover, the effect of degree of reheating on exergetic exergoeconomic parameters has been investigated. The results revealed that there is a specific value of degree of reheating for which the exergetic efficiency would be on its lowest value. This point would be of importance during optimization procedure of reheating direct steam generation solar plants.

Umit Unver et al. [7] Cost analysis has a significant importance to obtain the optimum marketing price of the product of thermal systems to maximize the benefit and/or minimize the cost. Thus, this paper focuses on the investigation of the magnitude of the change in costs with respect to load and environmental temperature variations. To achieve the objective, a useful and simple second law based thermo-economic model with instant access to production costs is introduced and generalized. The presented exergy costing method indicates that the cost of reversible power is the theoretically minimum cost, where the reversible power is the theoretical maximum power that can be gained from a thermal system. The analysis has been applied to a combined cycle power plant, which is located in Bursa/Turkey. The effects of load and environmental temperature variations on costs are discussed and presented. Without considering the load effect, the cost of net electric power varied from 29 to 32\$$MW^{-1}$ h^{-1}, and about 40–45% of the cost of net electric power is composed of cost of irreversibility, while its 55–60% of it is the cost of reversible power. It is shown that the augmentation in the costs are not continuous with the environmental temperature decrease. In addition, there is an extremum at the temperature range between 5 and 108^0 C.

Lalatendu Pattanayak [8] shows an exergy analysis of 88.71 MW 13D2 gas turbine (GT) topping cycle is carried out. Exergy analysis based on second law was applied to the gas cycle and individual components through a modeling approach. The analysis shows that the highest exergy destruction occurs in the combustion chamber (CC). In addition, the effects of the gas turbine load and performance variations with ambient temperature, compression ratio and turbine inlet temperature (TIT) are investigated to analyse the change in system behavior. The analysis shows that the gas turbine is significantly affected by the ambient temperature and with increase there is decrease in GT power output. The results of the load variation of the gas turbine show that a reduction in gas turbine load results in a decrease in the exergy efficiency of the cycle as well as all the components. The compressor has the largest exergy efficiency of 92.84% compared to the other component of the GT and combustion chamber is the highest source of exergy destruction of 109.89 MW at 100 % load condition. With increase in ambient temperature both exergy destruction rate and exergy efficiency decreases.

V. Tara Chand et al. [9] shows the exergy analysis is carried out by conducting mass, energy and exergy balance of each component in the gas turbine plant. Parametric analysis of the influence of various factors namely compression ratio (rp), compressor inlet air temperature (AT) and turbine inlet temperature (TIT) on irreversibilities of each and every component of gas turbine plant is carried out. Exergy analysis revealed that most sensitive components in gas turbine plant were combustion chamber.

Ravinder Kumar et al.[10] presents the thermal and economic performance of a 210 MW coal-fired power plant situated in North India. Analysis is used to predict coal consumption rate, overall thermal efficiency, mass flow rate of steam through boiler, and Net present value (NPV) of plant for given load. Thermodynamic analysis was carried out using mass and energy equations followed by empirical correlations. Predicted mass flow rate of steam, coal consumption rate, and thermal efficiency give fair agreement with plant operating data. The economic analysis includes operational activities such as equipment cost, fuel cost, operations and maintenance cost, revenue, and plant net present value. From economic point of view, the effect of condensate extraction pump redundancy on net present value is observed to be sensitive than boiler feed pump redundancy.

2.2 Conclusion from literature review:

The effects of the inlet temperature of the gas turbine, the excess air factor, and the compressor pressure ratio on the performance of the supercharged boiler combined cycle (SBCC) were studied.). Energy losses mainly occurred in the condenser where (404.653 MW at Max load, 306.747 MW at 75% load and 278.849 MW at 50% load) is lost to the environment. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the Turbine system (42% at Max load, 59% at 75% load and 46.1 at 50% load) followed by the condenser (28% at Max load, 20.3% at 75% load) while at 50% load the feed water heaters showed more exergy destruction (27.7%) than condenser (23.8) and then the feed water heaters (20.8% at max load, 12.1% at 75% load) . In addition, the calculated thermal efficiency based on the specific heat input to the steam was 43% while the exergy efficiency of the power cycle was (44% - 48%). The exergy efficiency obtained for the heat exchanger, the steam turbine generator, the deaerator and the blower are 88%, 74%, 72% and 66% respectively. The effects of High Pressure steam temperature and pressure on the steam turbine generator energy and exergy efficiencies are investigated**.**

2.3 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.

2.4 Objective:

Detailed analysis need to be done to identify the loss value (INR) due to exergy destruction in the different component of the thermal power plant so that the critical component can be analysed to save the exergy destruction.

CHAPTER-3

EXERGY ANALYSIS

Exergy or, availability of a system signifies the part of the system energy that can be converted into maximum useful or desired work with respect to immediate surrounding condition, referred to as ‘dead state’. The term Exergy was used for the first time by Rant, Z. in 1956. According to 1^{st} law of thermodynamics energy is conserved, but energy is conserved only quantitatively and not qualitatively. But exergy is a manifestation of quality of energy and unlike energy, exergy is not conserved and in fact, get destroyed due to irreversibility during a process. The energy efficiency or, 1st law efficiency merely implies the ratio of desired or, useful work output against total energy input for a system. But this does not consider the true capability of the system by considering the thermodynamic limitations for which the system is not responsible. Efficiency based on exergy for any system like whole thermal power plant or, say, a turbine reflect the true capability of the concerned system i.e., their actual working capability against their maximum possible capability due to irreversibility present in the system process. Thus unlike, energy efficiency, exergy efficiency gives more insight into the problem and help design, analyses and performance improvement of the energy conversion systems more effectively by identifying the locations and associated irreversibilities. Many researchers are using exergy methodology for analyzing different types of energy conversion systems.

3.1. Energy and Exergy equations of steam power plant:

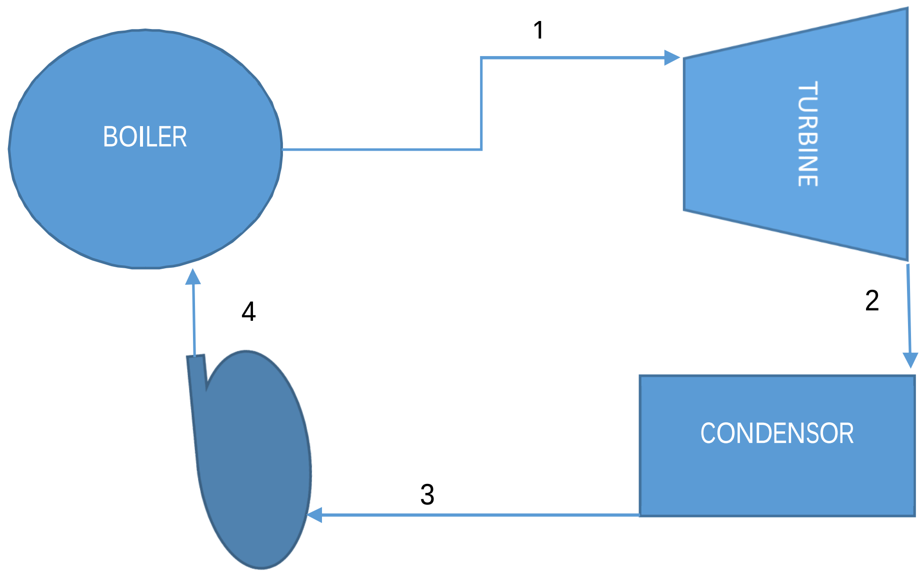


Fig.3.1 Steam power plant cycle

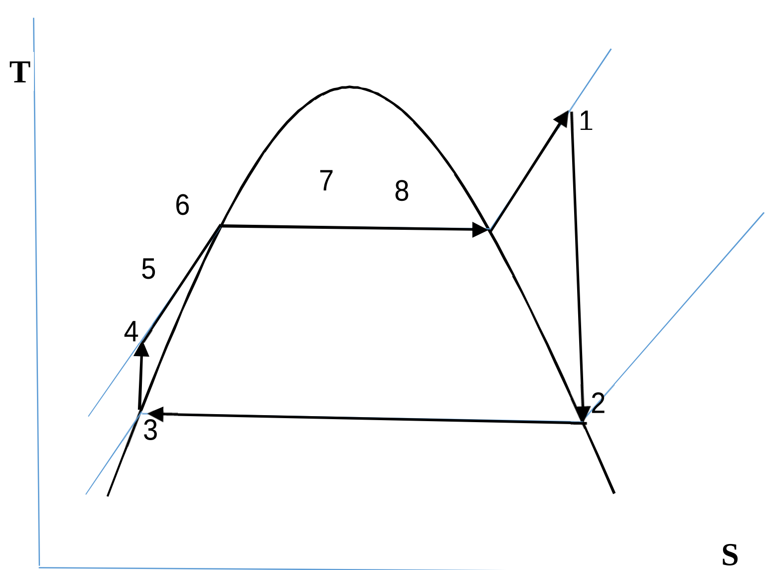


Fig.3.2 T-S diagram of steam cycle

3.1.1. Boiler:

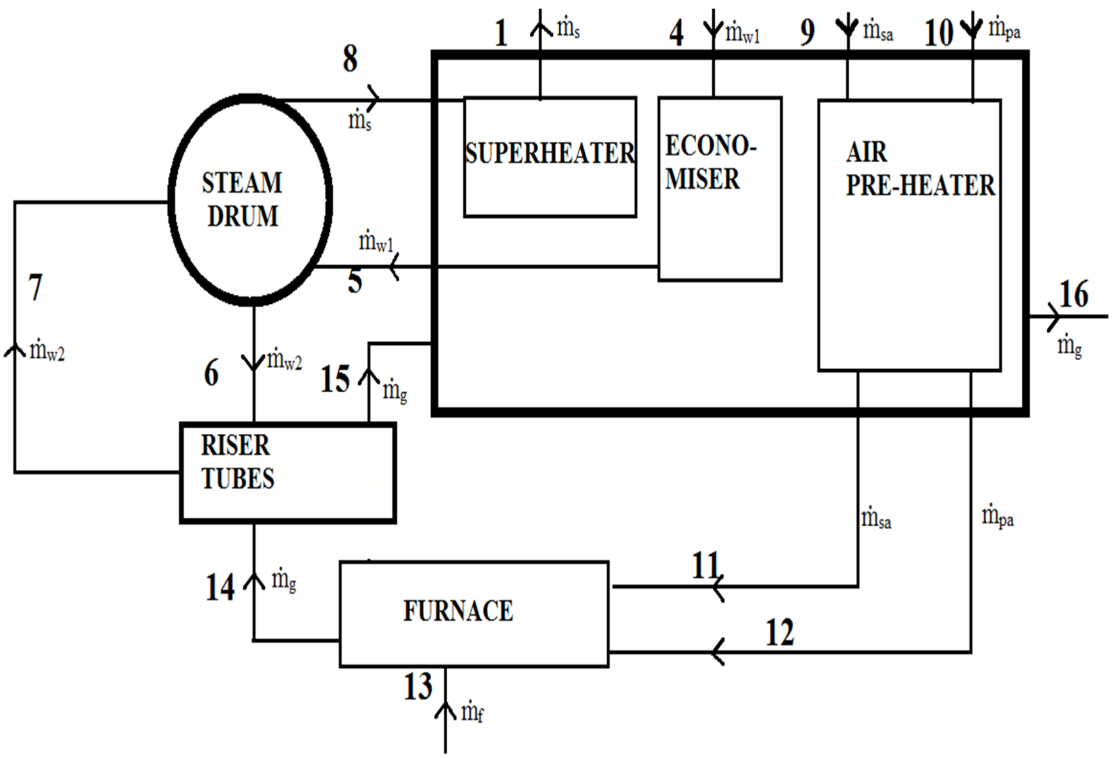


Fig. 3.3 Schematic representation of Boiler.

(a) Furnace:

Energy analysis:-

$$E_i = \dot{m}_f h_f + \dot{m}_{pa} h_{pa} + \dot{m}_{sa} h_{sa}$$

$$E_o = \dot{m}_g h_g$$

$$E_{loss} = E_i - E_o$$

$$\eta_I = E_o/E_i$$

Exergy analysis:-

$$X_i = \varepsilon_f + \varepsilon_{pa} + \varepsilon_{sa}$$

$$X_o = \varepsilon_g$$

$$I = X_i - X_o$$

$$\eta_{II} = X_o/X_i$$

Where-

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

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

$$\mathcal{E}_{g} = \dot{m}_{g}\,(h_{g} - T_o s_{g})$$

Exergy of fuel is given by the equation proposed by Shieh and Fan for calculating the Exergy of fuel

$$\mathcal{E}_f = 34183.16(C) + 21.95\,(N) + 11659.9\,(H) + 18242.90\,(S) + 13265.9\,(O)\ KW$$

(b). Heat Recovery System-

Energy analysis:-

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

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

$$E_o = \dot{m}_{sup}\,(h_o - h_i)_{sup} + \dot{m}_w\,(h_o - h_i)_w + \dot{m}_{pa}\,(h_o - h_i)_{pa} + \dot{m}_{sa}\,(h_o - h_i)_{sa}$$

$$E_{loss} = E_i - E_o$$

$$\eta_I = E_o/E_i$$

Exergy Analysis:-

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

$$X_o = (\mathcal{E}_{sup\,(o)} - \mathcal{E}_{sup\,(i)}) + (\mathcal{E}_{w\,(o)} - \mathcal{E}_{w\,(i)}) + (\mathcal{E}_{pa\,(o)} - \mathcal{E}_{pa\,(i)}) + (\mathcal{E}_{sa\,(o)} - \mathcal{E}_{sa\,(i)})$$

$$X_o = [\dot{m}_{sup}\,(h_{sup} - T_o s_{sup})_o - \dot{m}_{sup}\,(h_{sup} - T_o s_{sup})_i] + [\dot{m}_w\,(h_w - T_o s_w)_o - \dot{m}_w\,(h_w - T_o s_w)_i] + [\dot{m}_{pa}\,(h_{pa} - T_o s_{pa})_o - \dot{m}_{pa}\,(h_{pa} - T_o s_{pa})_i] + [\dot{m}_{sa}\,(h_{sa} - T_o s_{sa})_o - \dot{m}_{sa}\,(h_{sa} - T_o s_{sa})_i]$$

$$I = X_i - X_o$$

$$\eta_{II} = X_o/X_i$$

3.1.2 Analysis of Turbine:

(a) Energy analysis:

For adiabatic turbine-

$$\mathbf{E_i = \dot{m}_s h_1}$$

$$\mathbf{E_o = \dot{m}_s h_2}$$

Energy balance

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

$$\mathbf{\eta_I = 1 - E_o / E_i}$$

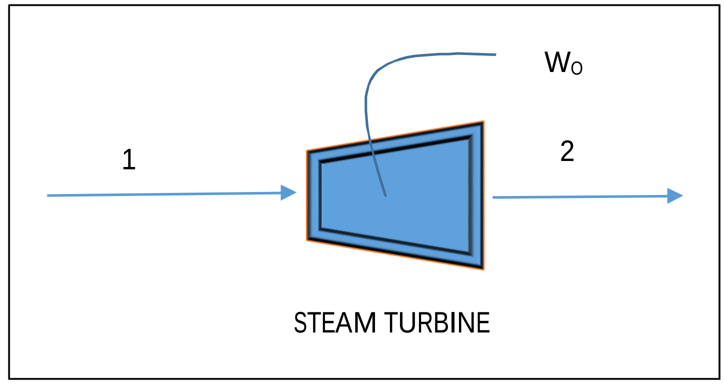


Fig.3.4 Steam Turbine

(b) Exergy analysis:

$$\mathbf{X_i = \dot{m}_s(h_1 - T_o s_1)}$$

$$\mathbf{X_0 = \dot{m}_s(h_2 - T_o s_2)}$$

Exergy balance

$$\mathbf{X_i - I - W_o = X_0}$$

$$\mathbf{\eta_{II} = 1 - I / (X_i - X_0)}$$

COMPONENTS	EXERGY DESTRUCTION	EXERGY EFFICIENCY
1. PUMP	$I_P = X_3 + W_P – X_4$	$\eta = \frac{X_4 – X_3}{W_P}$
2. CONDENSOR	$I_{con} = (X_2 – X_3) – (X_{3'} – X_{2'})$	$\eta = 1- \frac{Icon}{(X2 – X3)}$
3. TURBINE	$I_t = X_i – X_o – W_t$	$\eta = \frac{W_t}{(X_i - X_o)}$

Table3.1 Definitions of the exergy destruction rate and the exergy efficiency plant

3.2. Energy and Exergy analysis of Gas Power Plant:

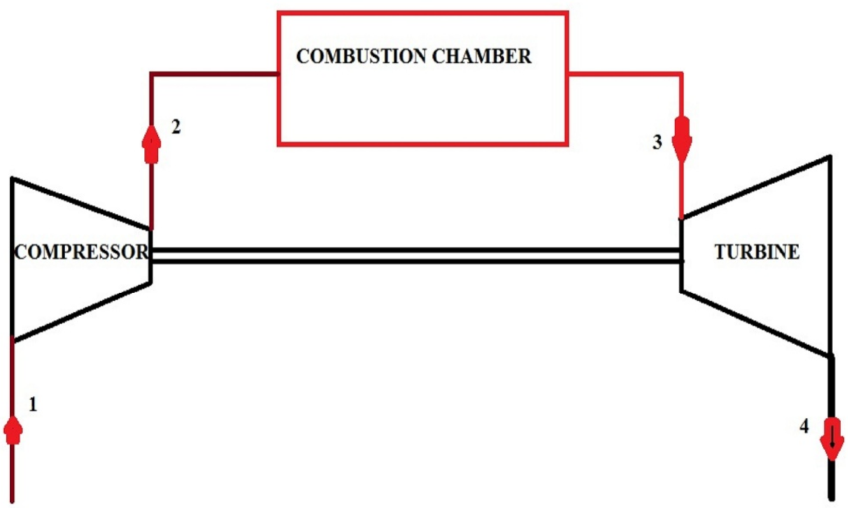


Fig.3.5 Typical Gas Power Plant Cycle.

Energy and Exergy analysis:

3.2.1 Compressor:

From the above figure-

Energy balance:

$\dot{m}_a h_1 = W_C + \dot{m}_a h_2$ (assuming isentropic compression with no change in K.E and P.E)

Exergy balance:

$$X_1 + W_C = X_2 + I_C$$

$$I_C = X_1 - X_2 + W_C$$

Where

$$X_1 = \dot{m}_a [h_1 - h_0 - T_0 (s_1 - s_0)]$$

$$X_2 = \dot{m}_a [h_2 - h_0 - T_0 (s_2 - s_0)]$$

3.2.2 Combustion Chamber:

Energy balance:

$$\dot{m}_a h_2 + \dot{m}_f \, LCV = (\dot{m}_a + \dot{m}_f) h_3$$

Where

LCV is the lower calorific value of the fuel

Exergy balance:

$$X_2 + X_f = X_3 + I_{cc}$$

$$X_f = X_{ph} + X_{ch}$$

$$X_{ph} = \dot{m}_a [h_f - h_0 + T_0 (s_f - s_0)]$$

$$X_{ch} = \dot{m}_f e_{ch}$$

Where

$$e_{ch} = x_i e_{chi} + R T_0 \Sigma x_i \mathrm{Loge} x_i + G_E$$

Where G_E is Gibbs free energy which is a negligible quantity in a gas mixture operated at low pressure. So for the calculation of fuel exergy, the given expression does not hold good. Thus, the fuel exergy can be calculated as the ratio of fuel exergy to lower heating value of fuel-

$$\Omega = \frac{ef}{LCV}$$

e_f is the specific exergy of the fuel.

For gaseous fuel with composition C_XH_Y, the value of Ω can be calculated as

$$\Omega = 1.033 + 0.0169 \frac{Y}{X} - \frac{0.0698}{X}$$

For NAPHTHELENE **($C_{10}H_8$) X=10, Y=8**

Then

$$\Omega = 1.00954$$

$$X_f = \dot{m}_f (1.00954 * LCV)$$

3.2.3 Gas Turbine:

Energy balance: assuming adiabatic turbine-

$$\dot{m}_a h_3 = W_C + \dot{m}_a h_4$$

Exergy balance:

$$X_3 = W_T + X_4 + I_t$$

Where

$$X_3 = (\dot{m}_a + \dot{m}_f)(h_3 - h_0 + T_0 (s_3 - s_0))$$

$$X_2 = (\dot{m}_a + \dot{m}_f)(h_2 - h_0 + T_0 (s_2 - s_0))$$

COMPONENTS	EXERGY DESTRUCTION	EXERGY EFFICIENCY
1. COMPRESSOR	$I_C = X_1 – X_2 + W_C$	$\eta = \frac{X_1 – X_2}{W_C}$
2. COMBUSTION CHAMBER	$I_{cc} = X_2 + X_f - X_3$	$\eta = \frac{X_3}{(X_2 + X_f)}$
3. GAS TURBINE	$I_T = W_T + X4 - X_3$	$\eta = \frac{W_T}{(X4 - X_3)}$

Table.3.2 Exergy destruction rate and exergy efficiency equations for GT components.

CHAPTER-4

EXERGY CONCEPT AND COST

4.1. Exergy analysis: 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 as will be explained later, and Exergy Losses relate to Lost Work.

One of the challenges in Thermodynamics compared to Mechanics is the introduction of somewhat abstract entities (or properties) describing P, V, T systems, such as Internal Energy, Entropy and Exergy. In addition, there are special energy functions such as Enthalpy, Helmholtz energy and Gibbs (free) energy that are important in thermodynamic analysis but can be difficult to fully comprehend. While Enthalpy is important for flow processes (open systems) in Mechanical Engineering Thermodynamics, Helmholtz energy (to define equations of state) and Gibbs free energy (for physical and chemical equilibrium) are important in Chemical Engineering Thermodynamics. Some text books introduce Internal Energy and Entropy as a way to be able to formulate the 1st and 2nd Laws of Thermodynamics:

• "Assuming there is a property called Internal Energy (symbol U or u in specific form); then the 1st Law of Thermodynamics can be formulated for closed and open systems".

• "Assuming there is a property called Entropy (symbol S or s in specific form); then the 2nd Law of Thermodynamics can be formulated for closed and open systems".

The dynamic Entropy balance for an open system in its most general form is then:

Fortunately, other text books, such as for example Moran and Shapiro □1□, make an effort trying to visualize what these properties really are. Common descriptions for Internal

Energy and Entropy are:

• Internal Energy (U) can be viewed as the Kinetic and Potential Energy at the micro level of the system; i.e. for atoms and molecules. Kinetic Energy at this level can take the form of translation, vibration and rotation of the molecules, while Potential Energy can be related to vibrational and electrical energy of atoms within molecules (vapor or liquid) or crystals (solids) as well as chemical bonds, electron orbits, etc.

• Entropy (S) can be viewed as a measure of the disorder (or “chaos”) in the system. More scientifically, it can be said to be a measure of the randomness of molecules in a system. In recent years, there has been a shift away from using the terms “order” and “disorder”, to using energy dispersion and the systems inability to do 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”.

Since the term “reversible processes” is used in the definition, one could simply say that:

• “The Exergy of a system at a certain thermodynamic state is the maximum amount of work that can be obtained when the system moves from that particular state to a state of equilibrium with the surroundings”.

Energy and Process Engineering Introduction to Exergy and Energy Quality This is the background why Exergy is related to Ideal Work. It should also be emphasized that there is a strong link between Exergy and Entropy since Entropy production (the term in the Entropy balance) is equivalent to Exergy Loss which again is equivalent to Lost Work. Also notice that while Exergy is the ability to produce work, Entropy was previously described as the systems inability to do work. Finally in this introduction, and as indicated in the very title of this document, Exergy is an indication of Energy Quality. Different energy forms have different quality (or different amounts of Exergy) in the sense that they have different capabilities to generate work. This is the main difference between the 1st and the 2nd Laws of Thermodynamics. The former states that energy is conserved and makes no distinction between energy forms, while the latter states that energy quality is destroyed, thus the different energy forms have different energy quality. The energy transformation processes in a system can only proceed from a higher quality form to a lower quality form unless there is some net input of energy quality (such as for example work) from the surroundings.

The Entropy Law itself emerges as the most economical in nature of all natural laws… the economic process and the Entropy Law is only an aspect of a more general fact, namely, that this law is the basis of the economy of life at all levels…" Might the justification of thermoeconomics be said in better words?

All real processes in a plant (or energy system) are nonreversible and some exergy is therefore destroyed, consuming forever some natural resources and creating a "cost." All natural resources have an economic cost: the more irreversible a process, the more natural resources are consumed (higher exergy cost) and the higher the required investment (higher thermoeconomic cost). If we can measure this thermodynamic cost by identifying, locating and quantifying the causes of inefficiencies in real processes, we can provide an objective economic basis by using the cost concept.

The search for the cost formation process is where physics connects best with economics, and thermoeconomics can be defined as a general theory of useful energy saving, where conservation is the cornerstone. Concepts such as thermodynamic cost, purpose, causation, resources, systems, efficiency, and structure and cost formation process are the bases of thermoeconomics. Thermoeconomics, term coined by Tribus and Evans, combines economic and thermodynamic analysis by applying the concept of cost (originally an economic property) to exergy (an energetic property), in order to, as stated by Bejan, Tsatsaronis and Moran, "provide the system designer or operator with information not available through conventional energy analysis and economic evaluations but crucial to the design and operation of a cost effective system."

4.2 The Cost Concept: The cost of a flow in a plant represents the external resources that have to be supplied to the overall system to produce this flow. Exergy cost accounting analysis distinguishes between exergy costs and monetary costs.

4.2.1 Exergy and Monetary Cost: The exergy cost of a mass and/or energy flow represents the units of exergy used to produce it, e.g., the exergy cost of the net power is the exergy provided by the natural gas to generate the electrical power delivered to the net by the cogeneration plant. These costs are a measure of the thermodynamic efficiency of the production process generating these flows. The unit exergy cost of a mass and/or energy flow represents the amount of resources required to obtain one unit of exergy. Thus, if the unit exergy cost of the electricity is three, it means that three units of plant exergy resources are consumed to obtain one exergy unit of electrical power.

The monetary cost takes into account the economic cost of the consumed fuel (i.e. its market price) as well as the cost of the installation and the operation of the plant, and it is defined as the amount of money consumed to generate a mass and/or energy flow. These costs could be considered as a measure of the economic efficiency of a process. Similarly to the unit exergy cost, the unit monetary cost (also called unit exergoeconomic cost or unit thermoeconomic cost) of a mass and/or energy flow is the amount of monetary units required to obtain one unit of exergy.

4.2.2 Average and Marginal Cost:

We can further distinguish between average costs, which are ratios and express the average amount of resources per unit of product, and marginal costs, which are a derivative and indicate the additional resources required to generate one more unit of the product under specified conditions.

4.2.3 NPV method for cost analysis:

The cost of electricity (COE) is defined as the net present value (INR) of the unit amount (kWh) of electricity generated by the plant cycle. In determining the COE, all costs over the lifetime are considered. These include the initial & capital investment and operating cost. The total capital investment includes the total direct plant cost and total indirect plant cost. Total direct plant cost involves the cost of equipment (i.e. boiler, steam turbine, condenser, generator and auxiliary equipment including condensate extraction pump, feed water pump, etc.) and costs associated with piping, electrical, civil works, direct installation cost, auxiliary services, instrumentation and controls, and site preparation. The total indirect plant cost includes the cost of engineering and setup and finally Operating cost (i.e. fuel cost, operations and maintenance cost).

Various methods have been developed to control the economics of thermal power plant like present worth method annual cost method, capitalized cost method and annual cost method. Peters et al. (1991) proposed that to account for the cost of the investment, the total capital cost must be placed on an annual basis (i.e. due to interest accumulated on the investment, depreciation, maintenance, insurance and taxes). There is always deterioration of equipment life and thus its depreciation cost loses value.

To have a realistic estimate of primary components or equipment, it is necessary to include the latest cost of these components. Thus, the data for costs for each Equipment / component of the plant in terms of power output capacity was obtained from literature search (Caputo et al. 2005; Hasler et al. 2009; Pauschert 2009; NETL 2012), and the cost of each component is fitted using power law in terms of installed capacity of the plant. The cost of the i^{th} component (C_i) is defined as

$$C_i = a_i\, MW^{b_i}$$

Where i is the steam turbine and Generator, steam boiler, boiler feed pump, condenser and condensate extraction pump respectively.

The values of empirical constants **a** and **b** listed in Table 4.1

S. no.	Equipment	a	b	References
1.	Steam turbine and generator	3501170.7	1.2	Caputo et al. (2005) Pauschert (2009)
2.	Steam boiler	76132335.1	0.8	Caputo et al. (2005) Hasler et al. (2009)
3.	Cond. extraction pump	9000	0.4	Caputo et al. (2005)
4.	Feed pumps	35000	0.6	Caputo et al. (2005)
5.	Condenser	17306123.8	0.5	Caputo et al. (2005) NETL (2012)
6.	Electrical works	59663488.1	0.6	Caputo et al. (2005) Pauschert (2009)
7.	Civil works	57,418,809.8	0.5	Caputo et al. (2005) Pauschert (2009)

Table 4.1.The values of empirical constants a and b.

The total direct plant cost includes the cost of equipment and other costs.

And can be written as

$$C_{direct} = C_{Equ.} + C_{other}$$

The cost of the equipment, can be written as

$$C_{Equ.} = \sum_i^n Ci + C_{piping} + C_{ashhandling} + C_{coalhandling}$$

The piping works cost (in INR) can be calculated using the polynomial as

$$C_{piping} = -895.91*(MW)^2 + 3674897.1*MW - 145238239.3$$

For the calculation of coal handling and ash handling cost, the following equation can be used

$$C_{coalhandling} = g_{conversion} * 96 * 10^5$$

$$C_{ashhandling} = g_{conversion} * 17 * 10^6$$

Other cost due to direct installation, instrumentation, auxiliary services, and control & site preparation can be grouped in another category as

$$C_{Other} = \mu C_{Equ.}$$

μ is a factor and have a value of **0.65**

$$C_{indirect} = \Upsilon\ C_{Equ.}$$

Here, Υ is a factor that accounts for engineering and plant startup, which is fixed at 0.22 in the present calculations.

Thus, total capital investment cost can be defined as the sum of total direct and total indirect plant cost as

$$C_{tci} = C_{direct} + C_{indirect}$$

The operating cost (includes the maintenance cost like labour cost, insurance cost, cost of power associated with boiler feed water pumps, purchasing cost of coal feedstock and condensate extraction pumps to run the thermal power plant) is taken to be paid yearly over the lifespan of coal fired power plant. It is likely to be changed in an economic climate (i.e. due to current rate of interest and escalation rate in the prices of coal, labour, maintenance, insurance and pumping power). Therefore, the influence of interest rate and the escalation rate must be taken in to account for the total operating cost over plant lifespan, the PWF can be defined as

$$PWK_j = \frac{1}{(1+i)^\wedge j}$$

Where PWF is the present worth factor.

Thus, the lifetime cost of fuel or coal, maintenance, labour and insurance can be obtained in terms of present worth factor and the escalation rate as following

Fuel or Coal cost

$$C_{coal} = \sum_{j=1}^{n} PWF_j * m_{coal,\,j} * C_c\,(1+P)^{(j-1)}$$

Labour cost

$$C_{lab} = \sum_{j=1}^{n} PWF_j * N_L * C_s\,(1+Q)^{(j-1)}$$

Maintenance cost

$$C_{maint} = \sum_{j=1}^{n} PWF_j * 0.015 * C_{tci} (1+R)^{(j-1)}$$

Insurance cost

$$C_{ins} = \sum_{j=1}^{n} PWF_j * 0.01 * C_{tci} (1+S)^{(j-1)}$$

The escalation rate on insurance (S) is assumed to be zero.

Pumping cost

$$C_{pumping} = \sum_{j=1}^{n} PWF_j * 8760 Av_{overall} * \sum_{k=1}^{N} \left(\frac{m}{d}\right) \Delta P / \eta * C_{ep} (1+T)^{(j-1)}$$

Here $Av_{overall}$, N, m, η and ΔP are the overall availability, number of pumps, pump efficiency, mass flow rate and pressure drop through the k^{th} pump, respectively. Also, 'N_L ' is the number of personnel employed, 'C' is the cost in INR, while subscripts maint, ins, ep, c and lab correspond to maintenance, insurance, price of electricity (INR/MWhr), coal cost and average annual labour cost (individual), respectively. In the present case, η has been taken to be 90 %. The maintenance and insurance costs are taken to be 1.5 and 1 % of the total capital investment obtained from the literature. The coal storage, fuel handling system and fume treatment costs are neglected in the present work. The taxes and financial charges have been neglected in this work.

The lifetime cost of purchasing cost of coal feedstock, maintenance and labour, insurance and cost of power associated with boiler feedwater pumps and condensate extraction pumps for running the thermal power plant can be calculated as

$$C_o = C_{coal} + C_{maint} + C_{lab} + C_{ins} + C_{pumping}$$

Likewise, the revenue over life span can be obtained from the sale of electricity in terms of PWF as

$$R_{an} = g_{MW} \sum_{j=1}^{n} PWF_j * 8760 Av_{overall} * MW * C_{ep} (1+T)^{(j-1)}$$

Caputo et al. (2005) reported that 10 % of the total revenue is consumed in internal affairs of the plant, which includes the pumping cost itself. Since we have included the pumping cost in the operating cost itself, the necessary adjustment in the net electric output would be required. From a baseline run, it was observed that the pumping cost was hardly 1 % of the total revenue. Therefore, the value of f was fixed at 91 %.

For assessment of the economic effectiveness of the investments, the NPV method is most frequently used. In the present work, therefore, the NPV method was employed. The expression of the net present value of plant on lifetime basis can be written as

$$NPV_{lifetime} = R_{lifetime} - (C_o + C_{tci})_{lifetime}$$

Total personnel deployed in 210 MW unit	Total monthly salary of plant personnel (INR)	Average monthly salary (INR)
240	9,600,000	40,000

Table 4.2 The total number of people deputed in the plant along with their salary.

Some actual data

Interest rate **(i =9%)**

Cost of electricity **(C_{ep}= 4500 INR/KWh i.e 4.5sRs/unit)**

Plant life **(n=35)**

Parameter	Notation	Value	References
Fuel cost	C_c	2.7(INR/Kg)	
Average labour cost on annual basis	C_s	4,80,000(INR/year)	
Number of laborers employed	N_L	240	
Interest rate	i	9	
Factor to account for power	g_{mw}	0.9	Caputo et al. (2005)
Current price of electricity	C_{ep}	4500 (INR/MWh)	HERC (2013)

Table 4.3 shows the fuel cost, factor to account for expenses due to installation, instrumentation, engineering and plant startup, factor to account for power consumption within plant, interest rate and escalation rate have been represented.

Parameter	Notation	Escalation rate (%)	Reference
Current price of electricity	T	5	Ranganathan (2005)
Fuel cost	P	6.6	Saxena (2013)
Labour salary	Q	10	
Maintenance cost	R	7.4	HERC (2013)

Table 4.4. Escalation rates on various costs

To calculate the loss value due to exergy loss, mass of fuel consumption per hour is calculated for 210MW plant. Hence the cost of exergy can be calculated.

For 210MW plant, the coal consumption rate is 21Kg/sec.

Cost of the coal = 2.7 Rs/Kg

Cost of coal consumed per hour = 2.7*21*3600 = 204120 Rs

Exergy of the fuel = 29795 KJ/sec (as calculated in the chapter 3 for bituminous coal)

Exergy for an hour = 3600*29795 KJ

Cost exergy factor = 204120/107262000=$1.9\text{x}10^{-3}$

Using cost of exergy factor, the loss value (in Rs) can be calculated directly by multiplying this factor to exergy destruction for the different components. From the turbine output power, the cost associated with this power can be calculated and this is not loss value. This power is available to convert in to electrical power, which further transported to the consumer. This electrical power is set as the revenue for the power plant.

Direct & indirect cost, operational cost and other cost are shown in the above analysis. These cost added up and determine the cost of electrical power. Electrical power is sold as Rs/KWh (Rs/unit).

CHAPTER-5

RESULT AND DISCUSSION

After the analysis of different cycles on which the thermal power plant work with respect to energy, exergy and cost. The important points concluded, which shows the exergy and cost analysis are complimentary to each other. The exergy destruction shows a loss that can be recovered by using the suitable design of the various parts of the system and also it confirms the best possible operation of the power plant according to second law of Thermodynamics. As the exergy and cost of energy are complimentary to each other exergy destruction shows a loss, which can be quantify by analysis the system in mathematically. In the present work the analysis is done in the steam power plant.

5.1 Result from Energy and Exergy analysis:

- **5.1.1 Effect of A/F (air fuel ratio) at different inlet air pressure on-**
  - **Exergy of the fuel (e_g):** The variation of the e_g with A/F ratio is shown in Fig5.1 As the A/F ratio increases, exergy of the fuel decreases when the pressure of the primary & secondary air inducted through air preheater less than the 2bar (P_a). But as the pressure value increases above 2bar, exergy value starts increasing. Thus by increasing the inlet pressure of the air from air preheater exergy of the fuel can be achieved higher at lower value of the A/F ratio.
  - **Exergy destruction in the furnace:** The variation of the I_{fur} with A/F ratio is shown in Fig5.2. Exergy destruction will increase continuously on increasing the value of A/F ratio. At a particular value of A/F ratio on increasing the value of the air inlet pressure after the air preheater, exergy destruction rate also increases. Important point is that, at lower value of the P_a exergy destruction curve is less steep. The exergy destruction rate increases rapidly at higher air pressure.
  - **Second law efficiency:** The variation of the η_{2fur} with A/F ratio is shown in Fig5.5. Exergetic efficiency will decrease continuously on increasing the value of the air inlet pressure after the air preheater. Important point is that, at lower value of the P_a exergetic efficiency decreases. The exergetic efficiency decreases continuously at higher air pressure but increases at higher pressure at a particular A/F ratio.

- **Temperature of the hot gas:** The variation of the T_g with A/F ratio is shown in Fig5.7. Hot gas temperature will decrease continuously on increasing the value of A/F ratio. Important point is that, on increasing the inlet air pressure, there is no noticeable effect on the hot gas temperature.

➢ **5.1.2 Effect of A/F (air fuel ratio) at different inlet air temperature-**

- **Exergy of the fuel (e_g):** The variation of the e_g with A/F ratio is shown in Fig5.3. As the A/F ratio increases, exergy of the fuel decreases. And when the temperature of the primary & secondary air inducted through air preheater increases, the value of exergy increases for a particular value of A/F ratio.
- **Exergy destruction in the furnace:** The variation of the I_{fur} with A/F ratio is shown in Fig5.4. Exergy destruction will increase continuously on increasing the value of A/F ratio. At a particular value of A/F ratio on increasing the value of the air inlet temperature after the air preheater, exergy destruction rate also increases. Important point is that, at higher value of the T_a exergy destruction curve is less steep. The exergy destruction rate increases rapidly at lower air temperature.
- **Second law efficiency:** The variation of the η_{2fur} with A/F ratio is shown in Fig5.6. Exergetic efficiency will decrease continuously on increasing the value of the air inlet temperature after the air preheater. At a particular value of A/F ratio on increasing the value of the air inlet temperature after the air preheater, exergetic efficiency also increases.
- **Temperature of the hot gas:** The variation of the Tg with A/F ratio is shown in Fig5.8. Hot gas temperature will decrease continuously on increasing the value of A/f ratio. Important point is that, on increasing the inlet air temperature, the hot gas temperature increases slightly.

➢ **5.1.3 Effect of A/F (air fuel ratio) at different boiler pressure P_1 (bar) on-**

- **First law efficiency of HRS:** The variation of the η_{1hrs} with A/F ratio is shown in Fig5.9. First law efficiency increases as A/F ratio increases, but the variation is gradual. As the boiler pressure increases, the value of the first law efficiency increases.

- **Second law efficiency of HRS:** The variation of the η_{2hrs} with A/F ratio is shown in Fig5.11. Second law efficiency increases with increase in A/F ratio. On increasing the value of boiler pressure there is no change in the value of second law efficiency at a particular value of A/F ratio.
- **Exergy destruction in HRS:** The variation of the I_{hrs} with A/F ratio is shown in Fig5.14. I_{hrs} decreases with increase in A/F ratio. On increasing the value of boiler pressure there is no change in the value of I_{hrs} at a particular value of A/F ratio.

➢ **5.1.4 Effect of turbine inlet temperature T_1 at different boiler pressure P_1 (bar) on-**

- **Work output from the turbine:** The variation of the W_t with $T_1$is shown in Fig5.15. Work output increases with increase in T_1. On increasing the value of boiler pressure the value of work output at a particular value of $T_{1,}$ decreases.
- **First law efficiency of turbine:** The variation of the η_{t1} with $T_1$is shown in Fig5.16. First law efficiency increases with increase in T_1. On increasing the value of boiler pressure the value of first law efficiency at a particular value of T_1, decreases.
- **Second law efficiency of turbine:** The variation of the η_{2t} with T_1 is shown in Fig5.17. Second law efficiency increases with increase in T_1. On increasing the value of boiler pressure the value of second law efficiency at a particular value of T_1, decreases.
- **Exergy destruction in turbine:** The variation of the I_t with T_1 is shown in Fig5.18. Exergy destruction decreases with increase in T_1. On increasing the value of boiler pressure the value of exergy destruction at a particular value of T_1, increases.
- **Second law efficiency of condenser:** The variation of the η_{2cond} with T_1 is shown in Fig5.19. Second law efficiency increases with increase in T_1. On increasing the value of boiler pressure the value of second law efficiency at a particular value of T_1, decreases.

- **Exergy destruction in condenser:** The variation of the I_{cond} with T_1 is shown in Fig5.20. Exergy destruction decreases with increase in T_1. On increasing the value of boiler pressure the value of exergy destruction at a particular value of T_1, increases.
- **Mass flow rate of cold water through condenser:** The variation of the m_{cw} with T_1 is shown in Fig5.21. Mass flow rate of cooling water increases with increase in T_1. On increasing the value of boiler pressure the value of mass flow rate of water at a particular value of T_1, decreases.

➢ **5.1.5 Effect of A/F (air fuel ratio) at different turbine inlet temp. T_1 (bar) on-**

- **First law efficiency of HRS:** The variation of the η_{1hrs} with A/F ratio is shown in Fig5.10. First law efficiency increases as A/F ratio increases, but the variation is gradual. As the turbine inlet temp., the value of the first law efficiency increases.
- **Second law efficiency of HRS:** The variation of the η_{2hrs} with A/F ratio is shown in Fig5.12. Second law efficiency increases with increase in A/F ratio. On increasing the value of turbine inlet temperature the value of second law efficiency at a particular value of A/F ratio, increases.
- **Exergy destruction in HRS:** The variation of the I_{hrs} with A/F ratio is shown in Fig5.13. I_{hrs} decreases with increase in A/F ratio. On increasing the value of turbine inlet temperature the value of I_{hrs} at a particular value of A/F ratio, decreases.

5.2 Result from cost analysis:

In the present analysis, energy is quantified as cost and this cost is associated with the exergy loss or exergy destruction. Thus this cost shows a loss value and represented by C.

➢ **5.2.1 Effect of A/F ratio on loss value of-**

- **$C_{furnace}$ (INR):** The variation of the $C_{furnace}$ with A/F ratio is shown in Fig5.22. $C_{furnace}$ is the loss associated with the exergy destruction in the furnace. From the figure it can be seen that as the A/F ratio increases the loss value increases.

- **C_{hrs} (INR):** The variation of the C_{hrs} with A/F ratio is shown in Fig5.23. C_{hrs} is the loss associated with the exergy destruction in the heat recovery system. From the figure it can be seen that as the A/F ratio increases the loss value decreases in the heat recovery system.

- **5.2.2 Effect of turbine inlet temperature on loss value of-**
  - **C_t (NR):** The variation of the C_t with T_1 is shown in Fig5.24. C_t is the loss associated with the exergy destruction in turbine. From the figure it can be seen that as the value of T_1 increases the loss value decreases in the turbine but on increasing the boiler pressure loss value increases at a particular temperature.
  - **C_{cond} (INR):** The variation of the C_{cond} with T_1 is shown in Fig5.25. C_t is the loss associated with the exergy destruction in condenser. From the figure it can be seen that as the value of T_1 increases the loss value decreases in the condenser but on increasing the boiler pressure loss value increases at a particular temperature.
  - **C_{total} (INR):** The variation of the C_{total} with T_1 is shown in Fig5.26. C_t is the loss associated with the exergy destruction in furnace, heat recovery system, turbine and condenser. From the figure it can be seen that as the value of T_1 increases the loss value decreases but on increasing the boiler pressure loss value increases at a particular temperature.

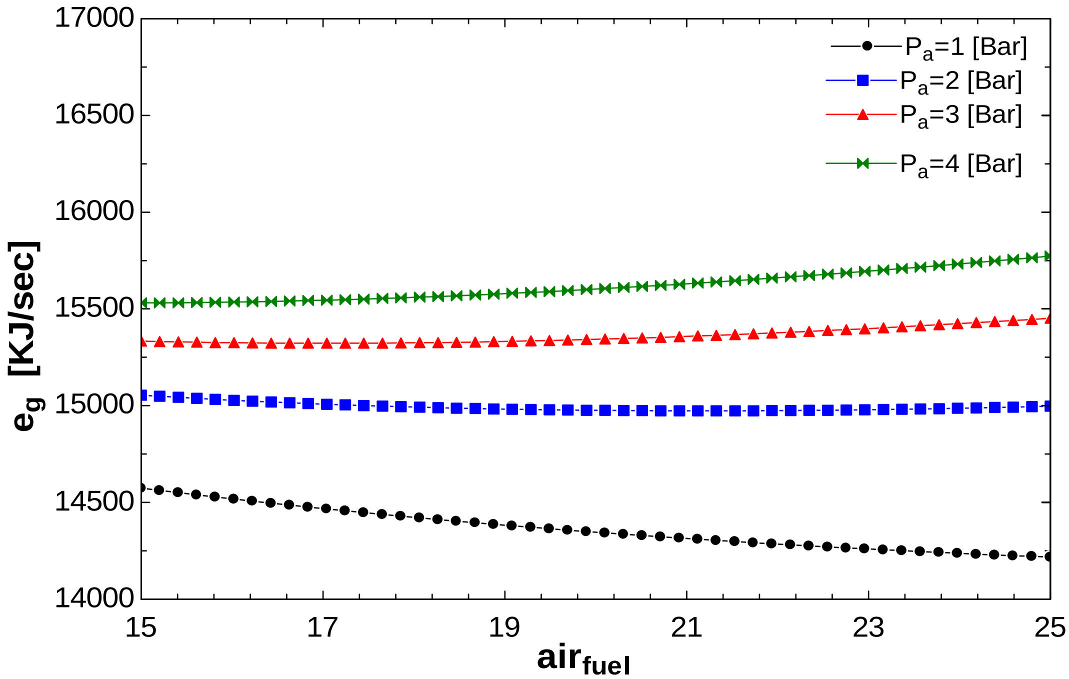


Fig.5.1 Variation of Exergy of hot gas with A/F ratio at different air pressure.

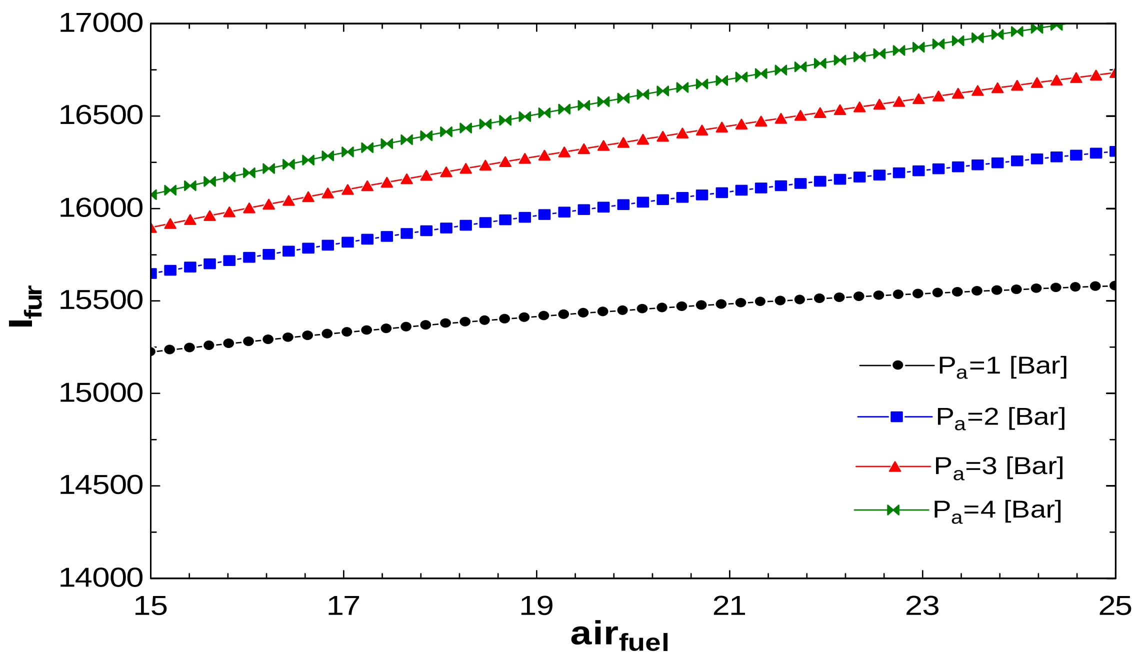


Fig. 5.2 Variation of Exergy destruction in furnace with A/F ratio at different air pressure.

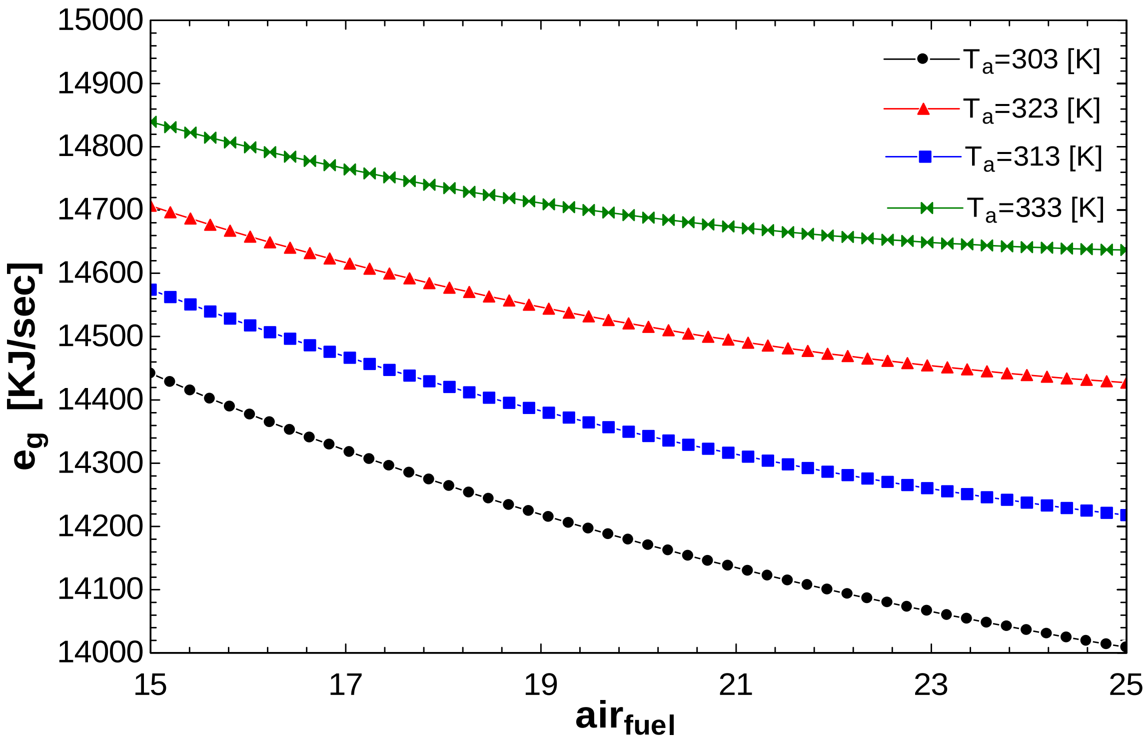


Fig. 5.3 Variation of Exergy of hot gas with A/F ratio at different air temperature.

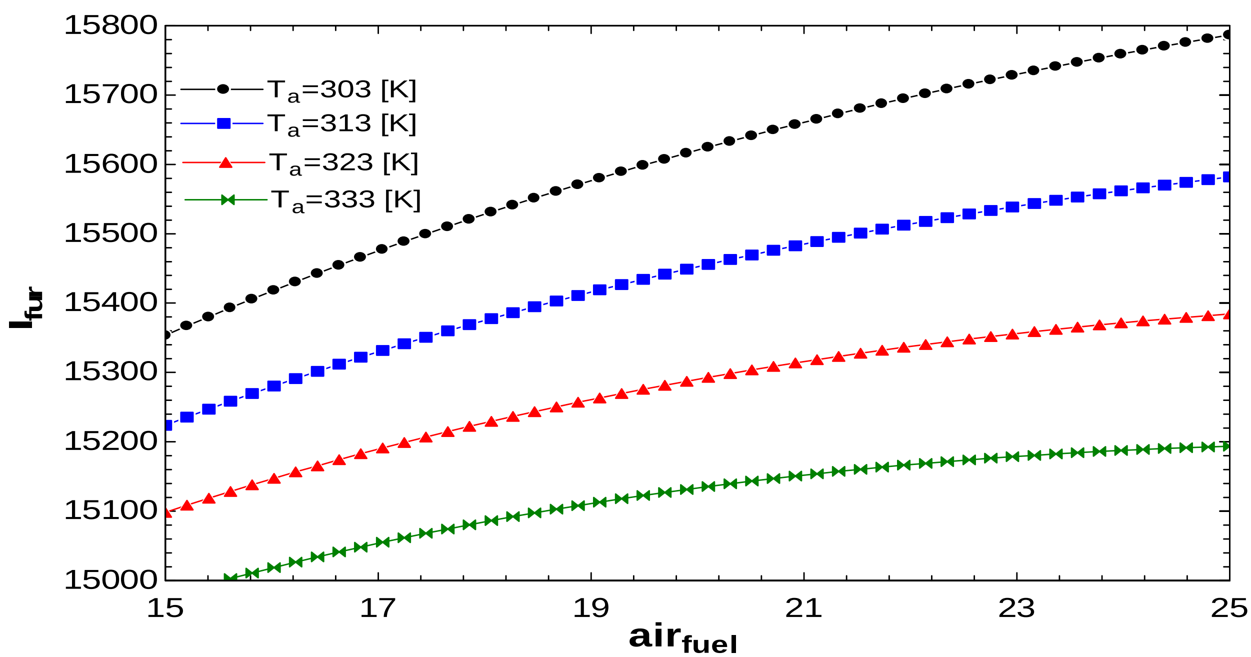


Fig. 5.4 Variation of Exergy destruction in furnace with A/F ratio at different air temperature.

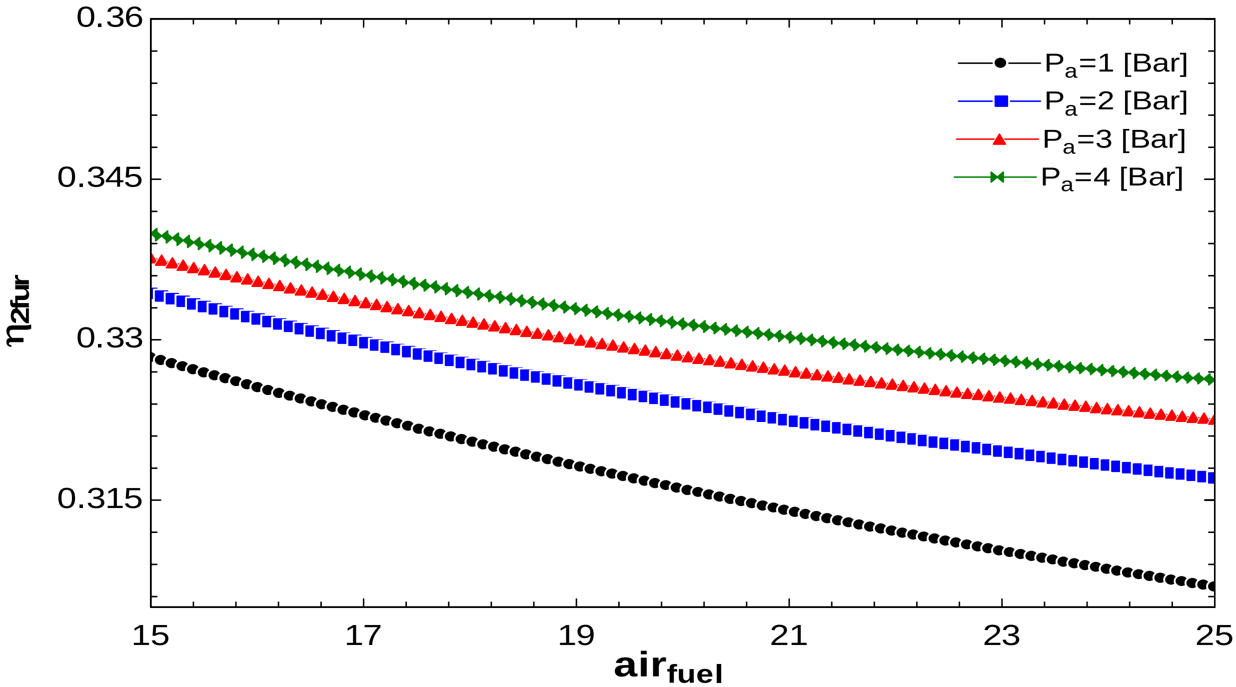


Fig. 5.5 Variation of Second law eff. of furnace with A/F ratio at different air pressure.

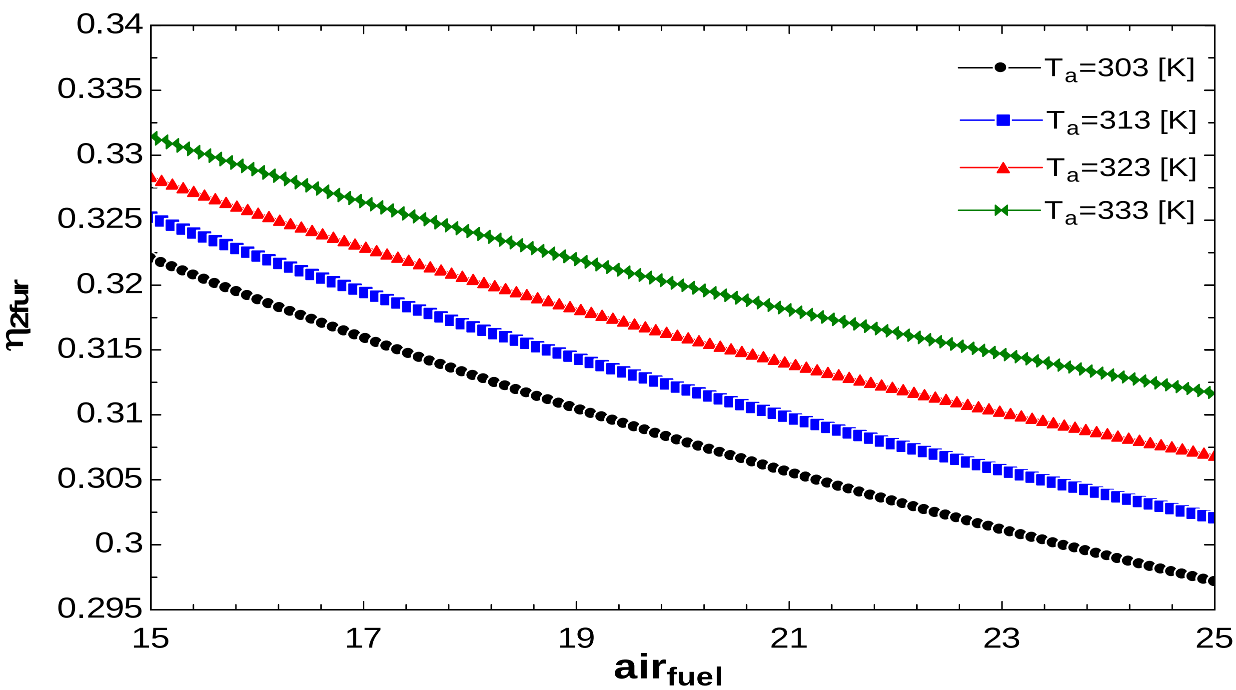


Fig.5.6 Variation of Second law eff. of furnace with A/F ratio at different air temperature.

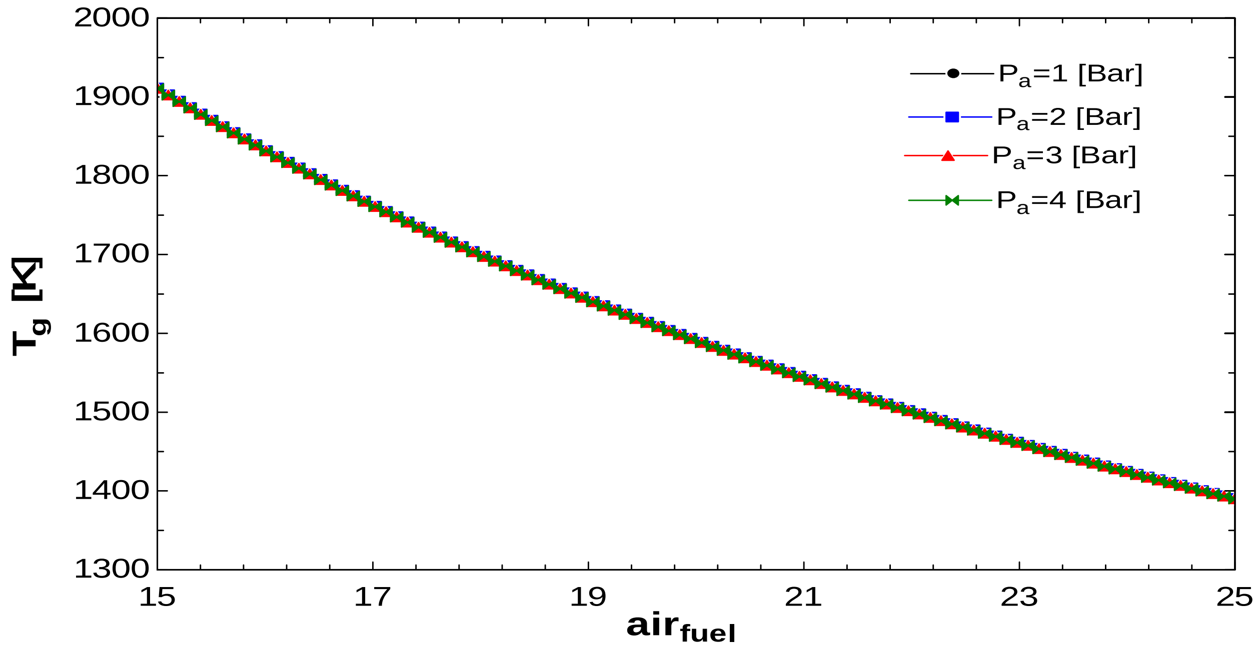


Fig. 5.7 Variation of hot gas temperature in furnace with A/F ratio at different air pressure.

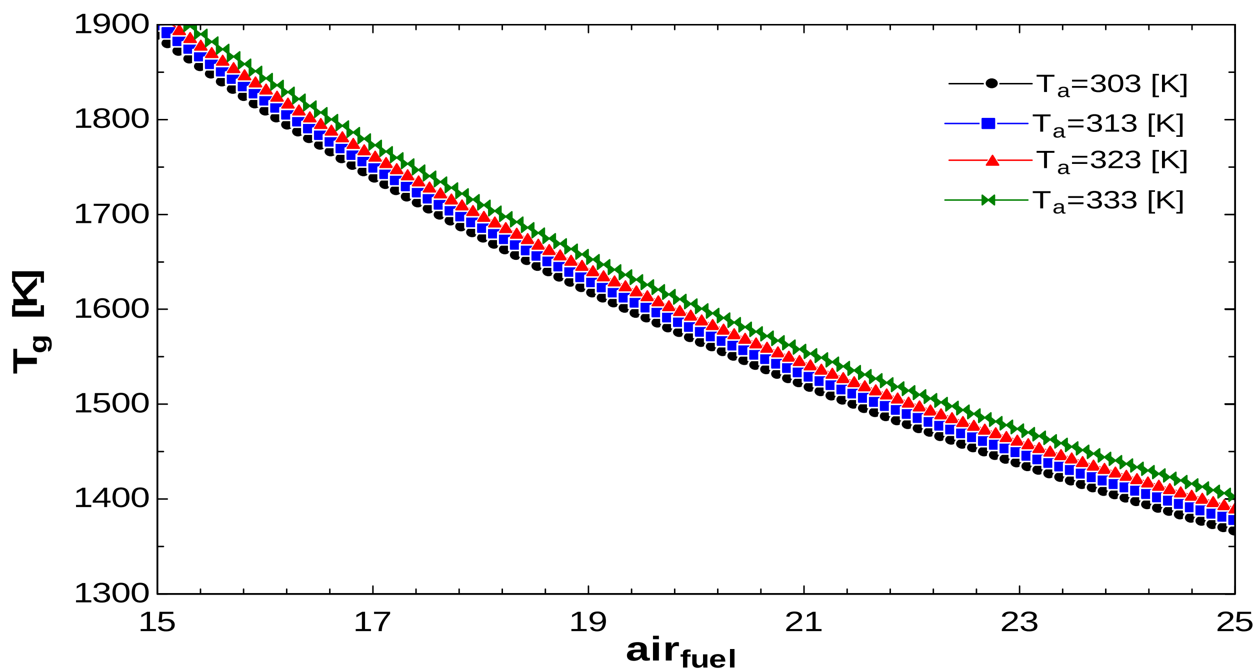


Fig. 5.8 Variation of hot gas temperature in furnace with A/F ratio at different air temperature.

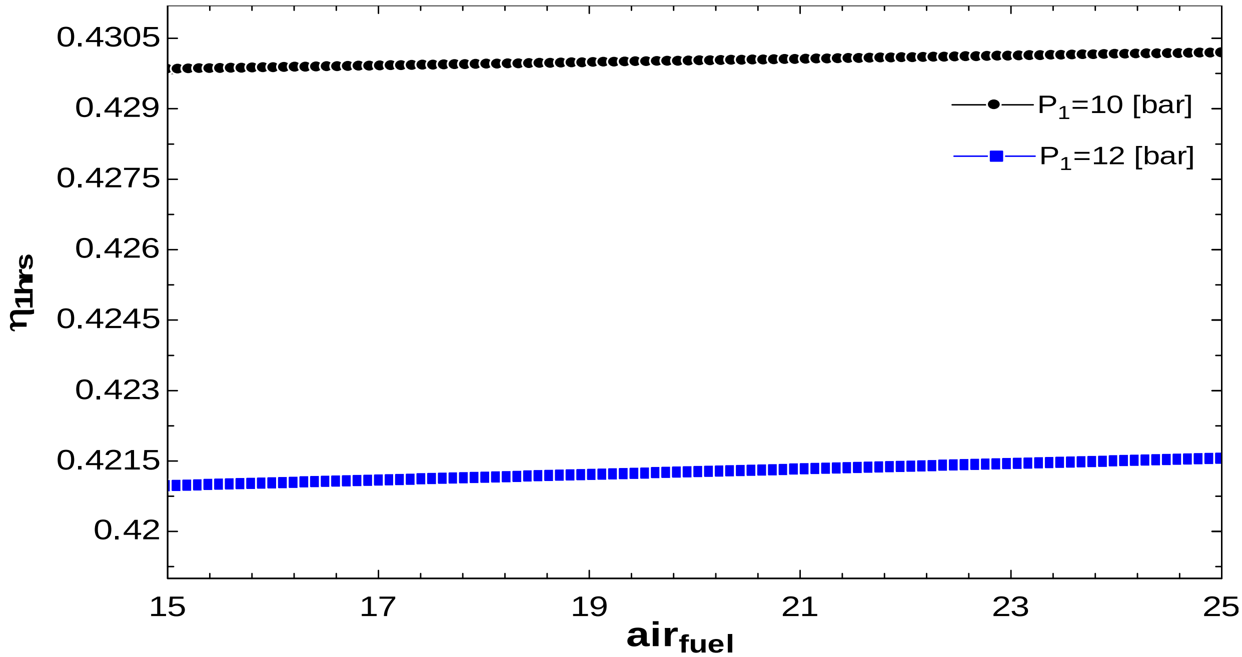


Fig. 5.9 Variation of first law eff. of HRS with A/F ratio at different boiler Pressure.

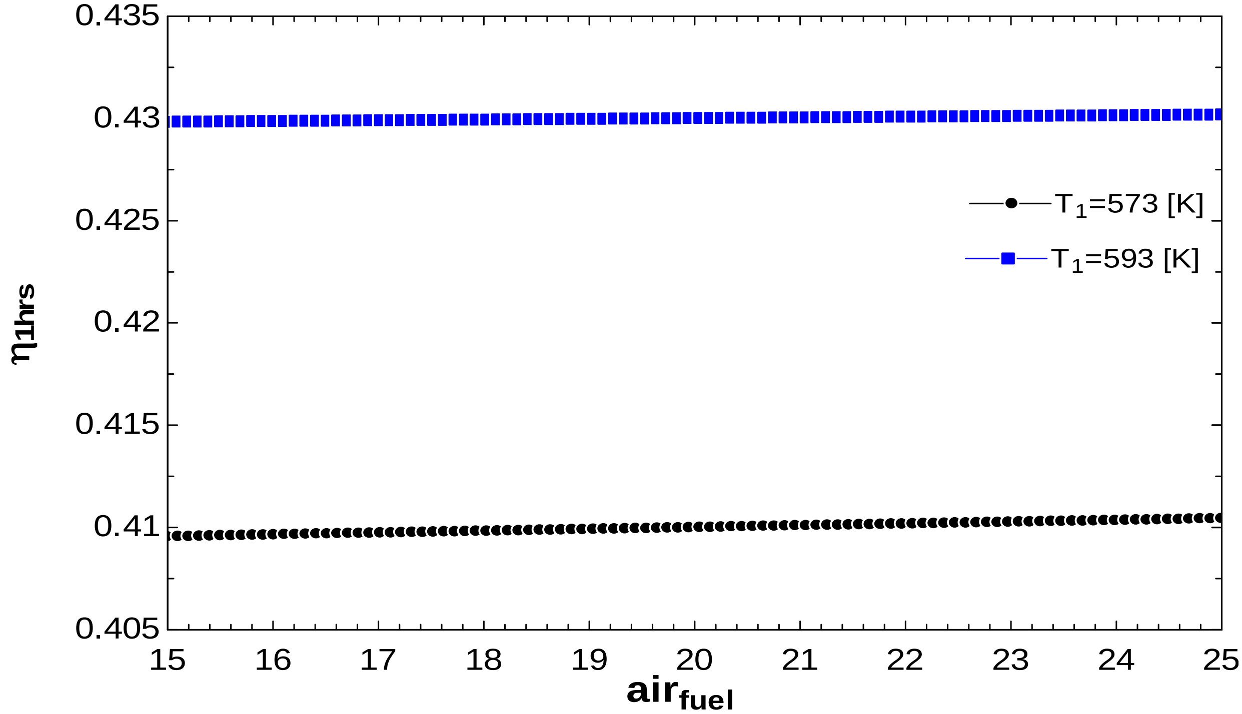


Fig. 5.10 Variation of first law eff. of HRS with A/F ratio at different steam Temperature.

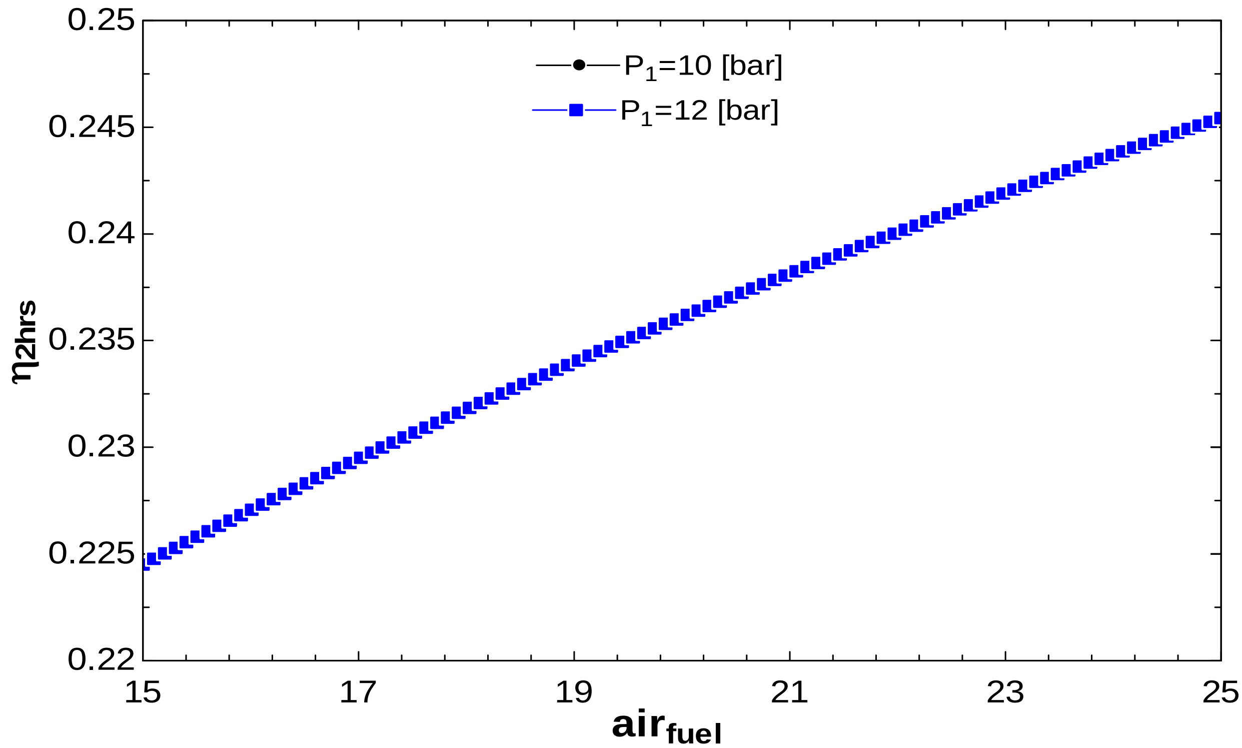


Fig. 5.11 Variation of second law eff. of HRS with A/F ratio at different boiler Pressure.

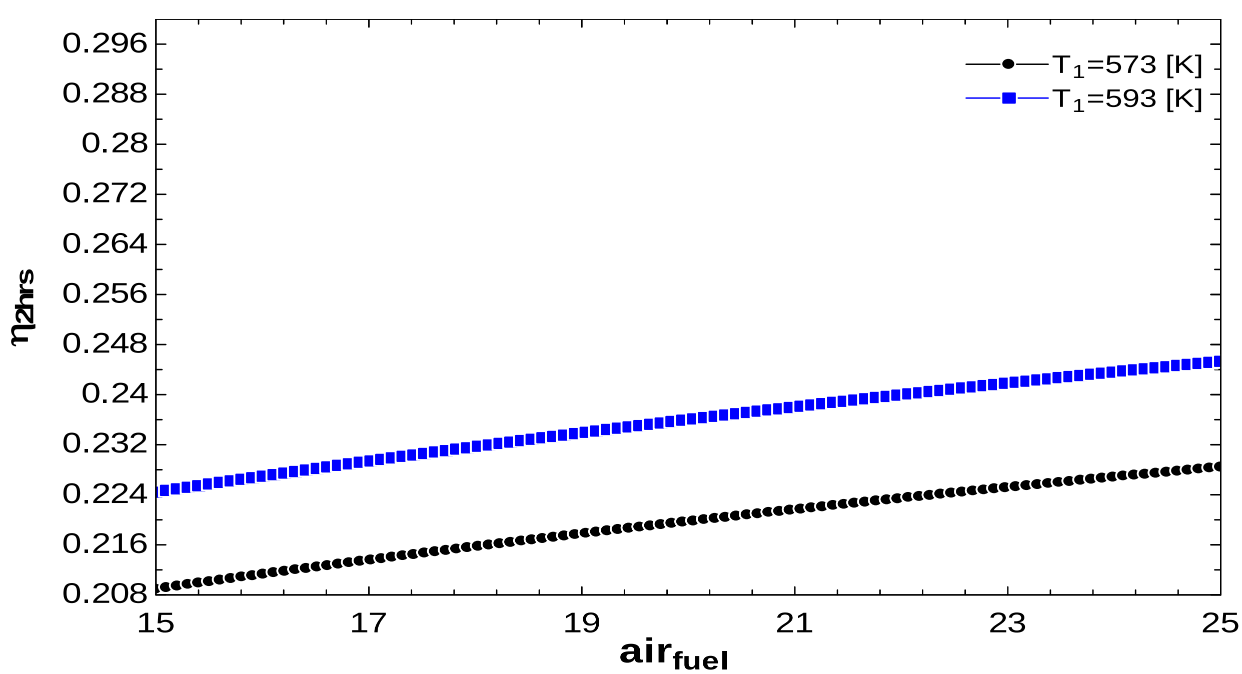


Fig. 5.12 Variation of second law eff. of HRS with A/F ratio at different steam Temperature.

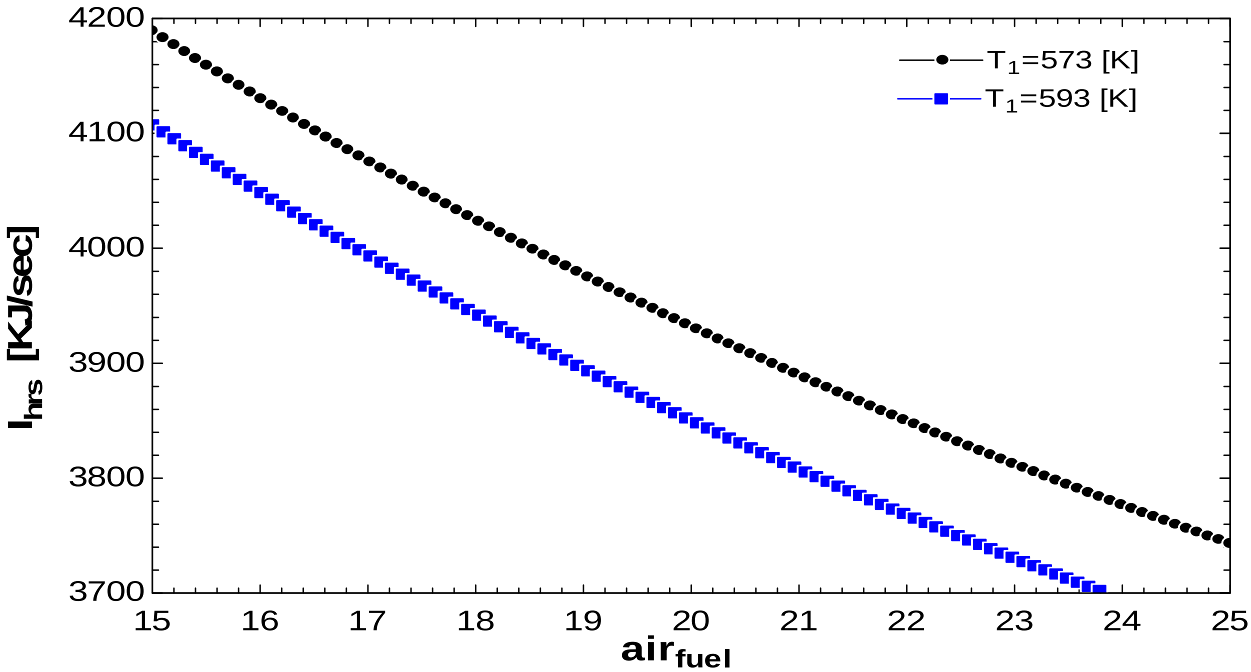


Fig. 5.13 Variation of Exergy destruction rate of HRS with A/F ratio at different steam Temperature.

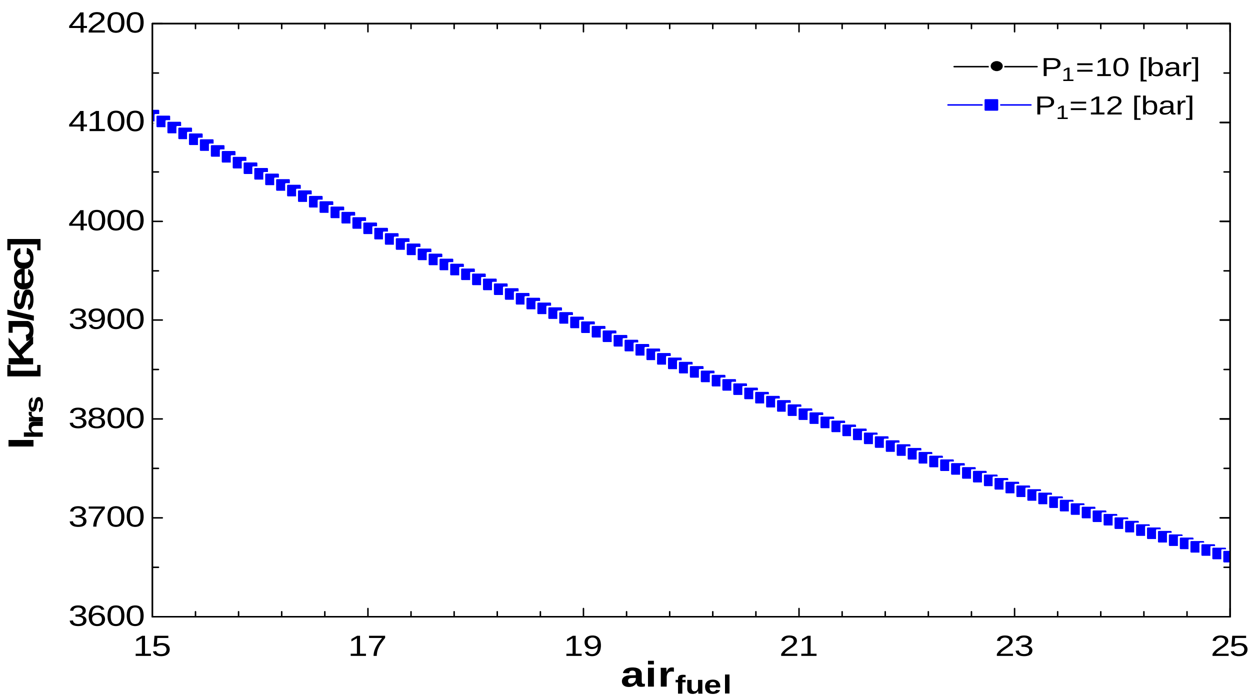


Fig. 5.14 Variation of Exergy destruction rate of HRS with A/F ratio at different boiler Pressure.

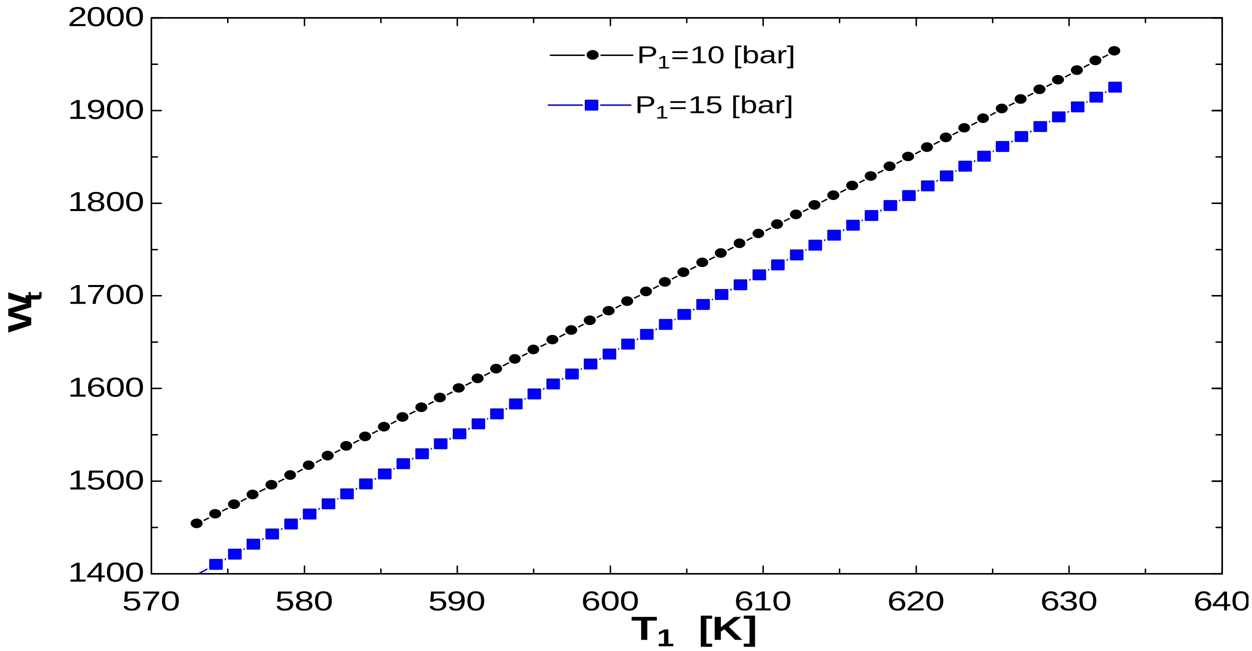


Fig. 5.15 Variation of Work output of turbine with T_1 at different boiler Pressure.

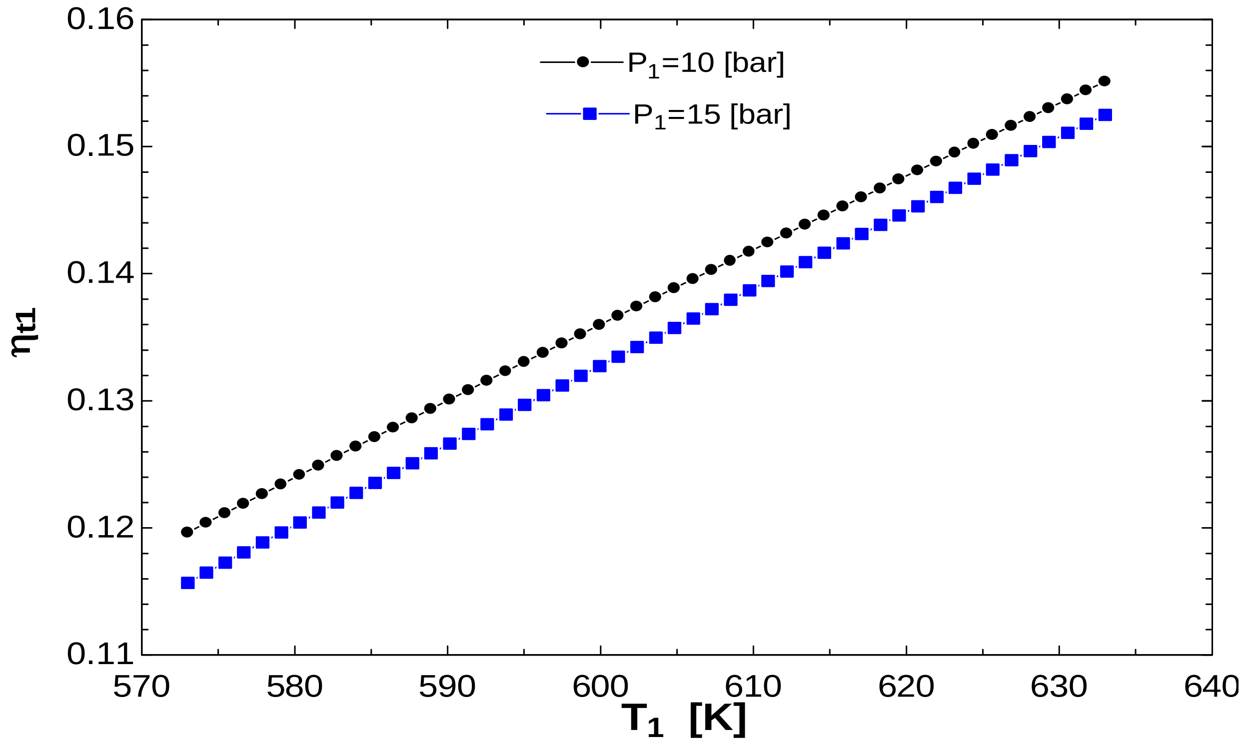


Fig. 5.16 Variation of first law eff. of turbine with T_1 at different boiler Pressure.

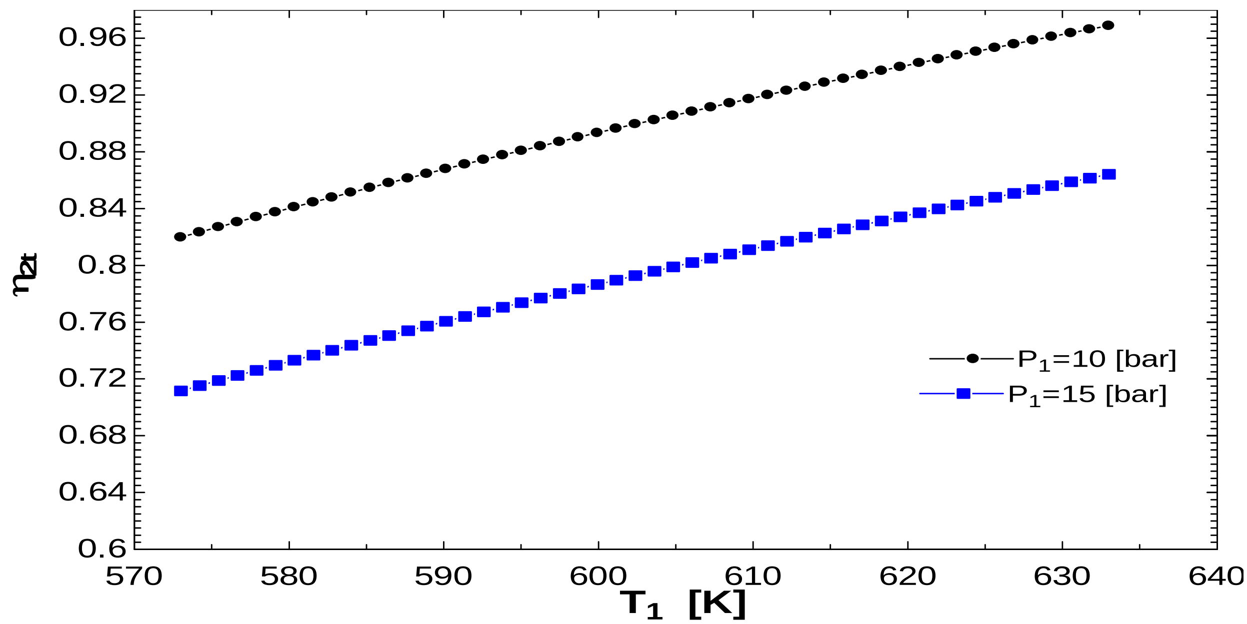


Fig. 5.17 Variation of Second law eff. of turbine with T_1 at different boiler Pressure.

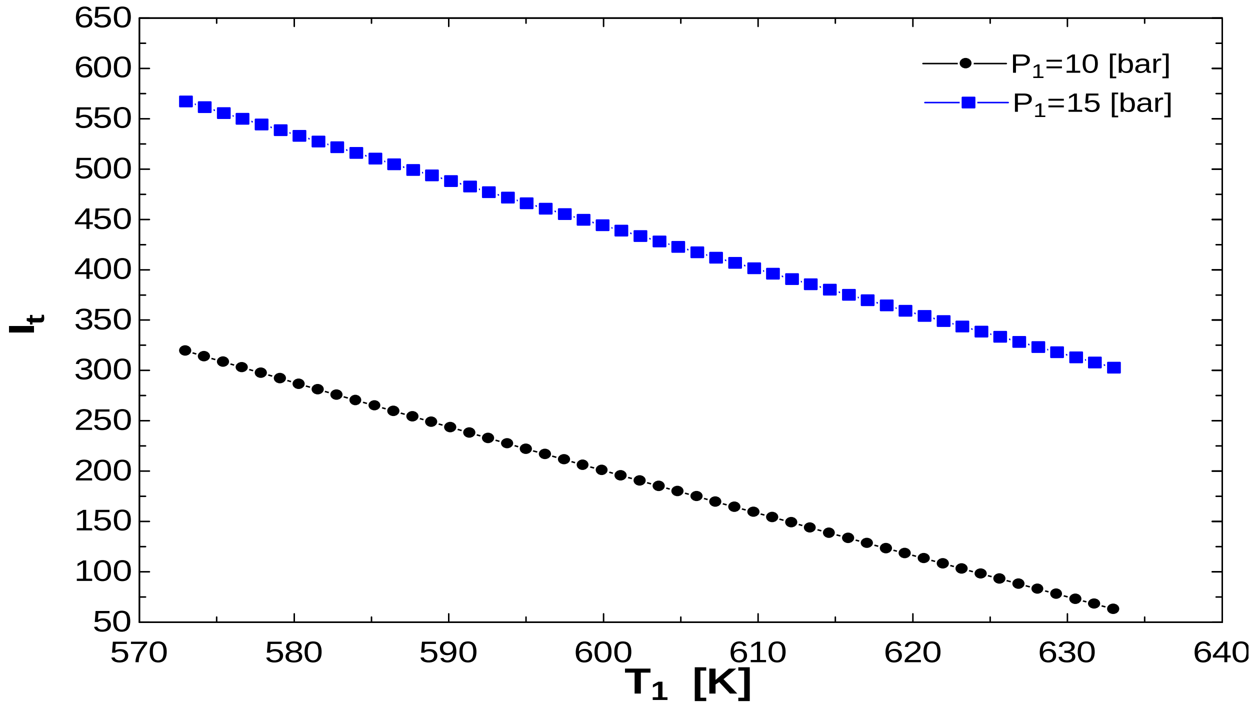


Fig. 5.18 Variation of Exergy destruction in turbine with T_1 at different boiler Pressure.

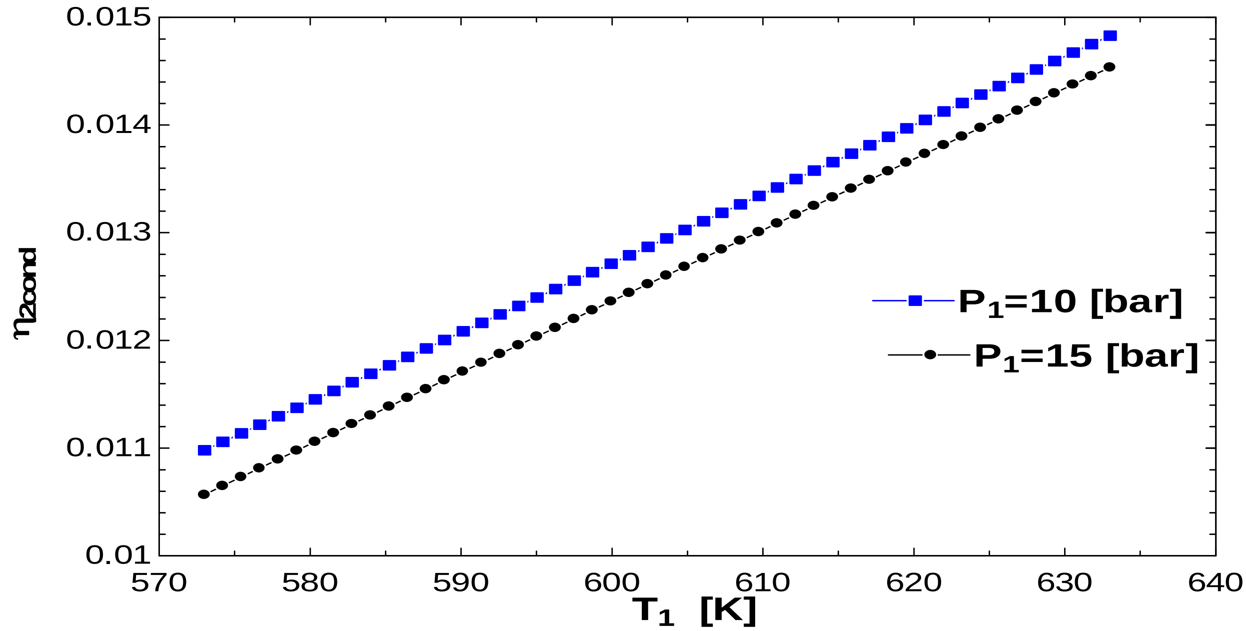


Fig. 5.19 Variation of Second law eff. of condenser with T_1 at different boiler Pressure.

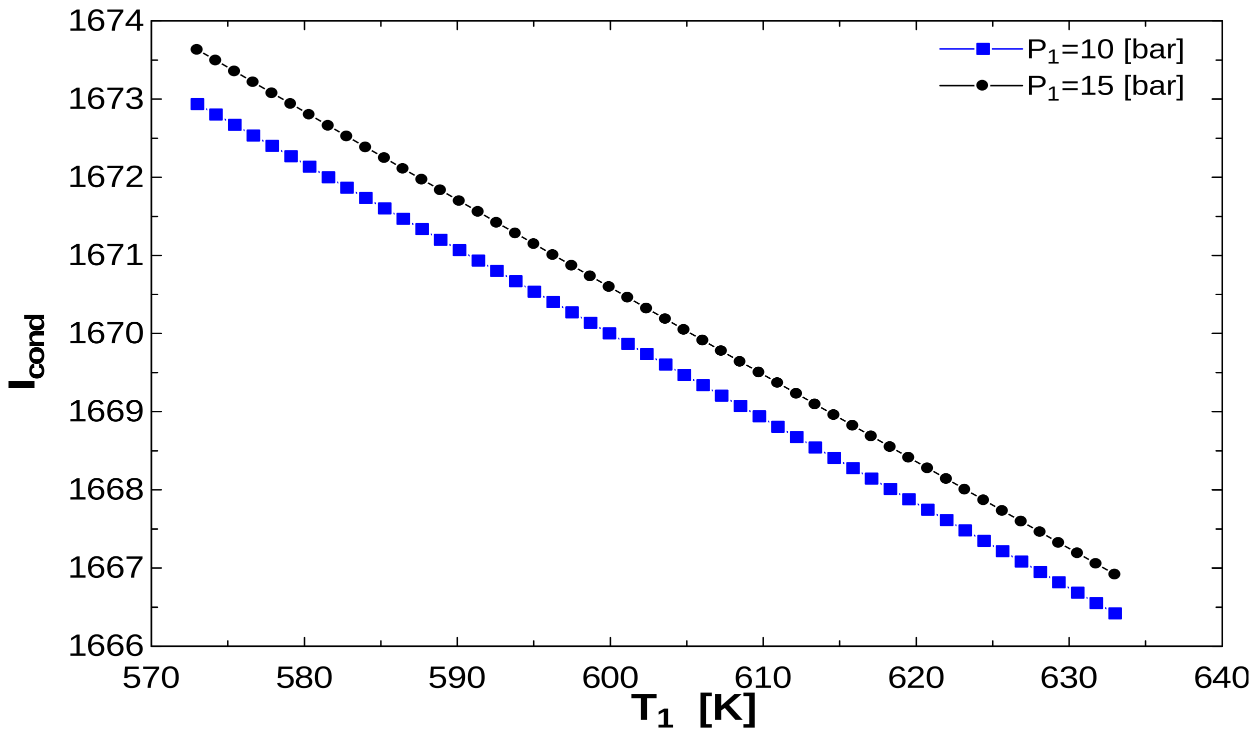


Fig. 5.20 Variation of Exergy destruction of condenser with T_1 at different boiler Pressure.

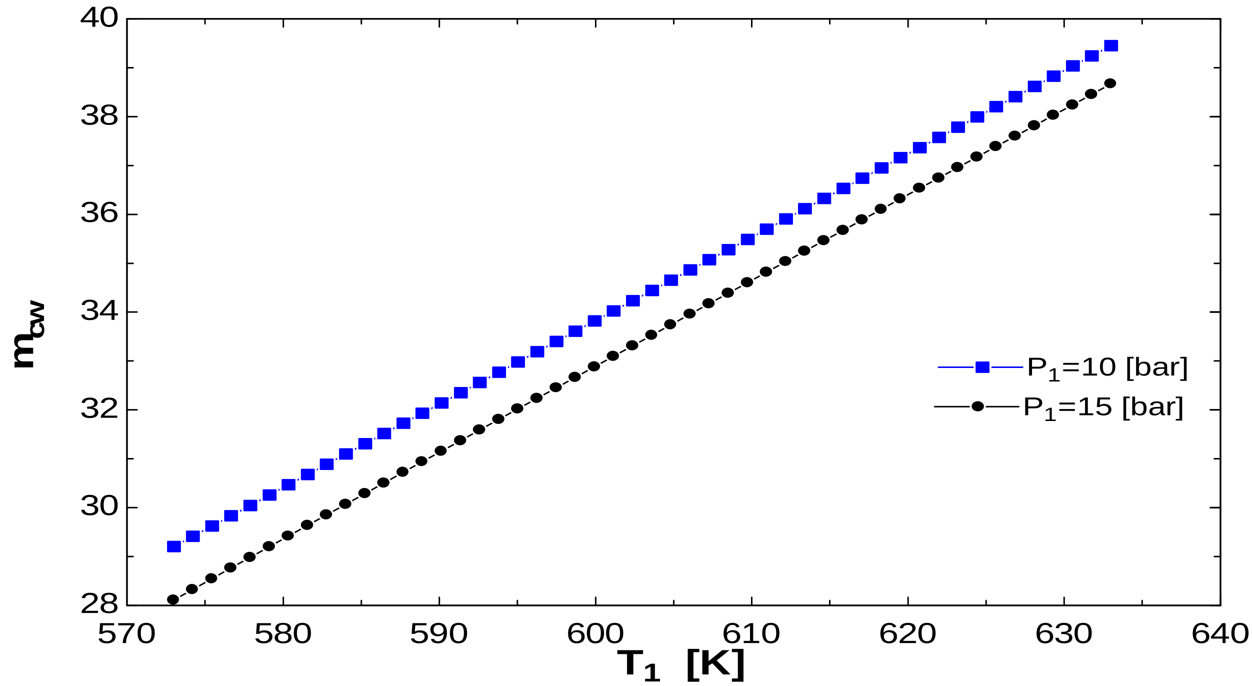


Fig. 5.21 Variation of mass flow rate of cold water through condenser with T_1 at different boiler Pressure.

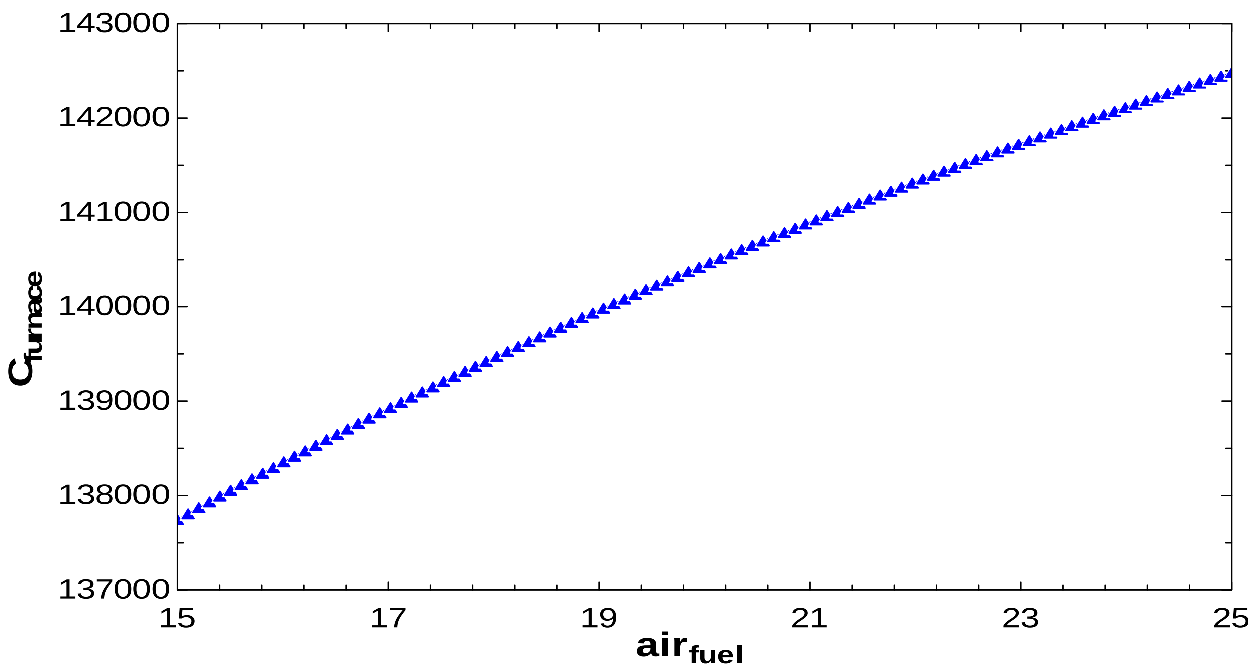


Fig. 5.22 Loss (INR) at furnace due to exergy destruction variation with A/F.

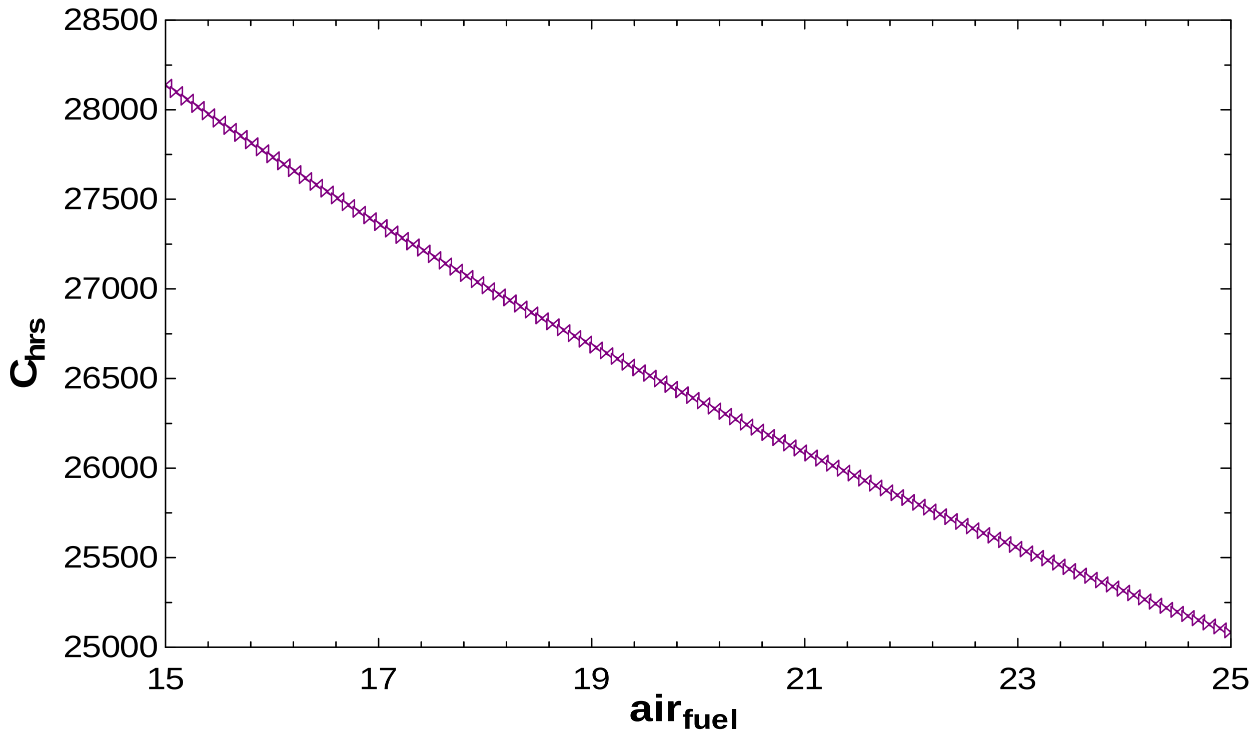


Fig. 5.23 Loss (INR) at HRS due to exergy destruction variation with A/F.

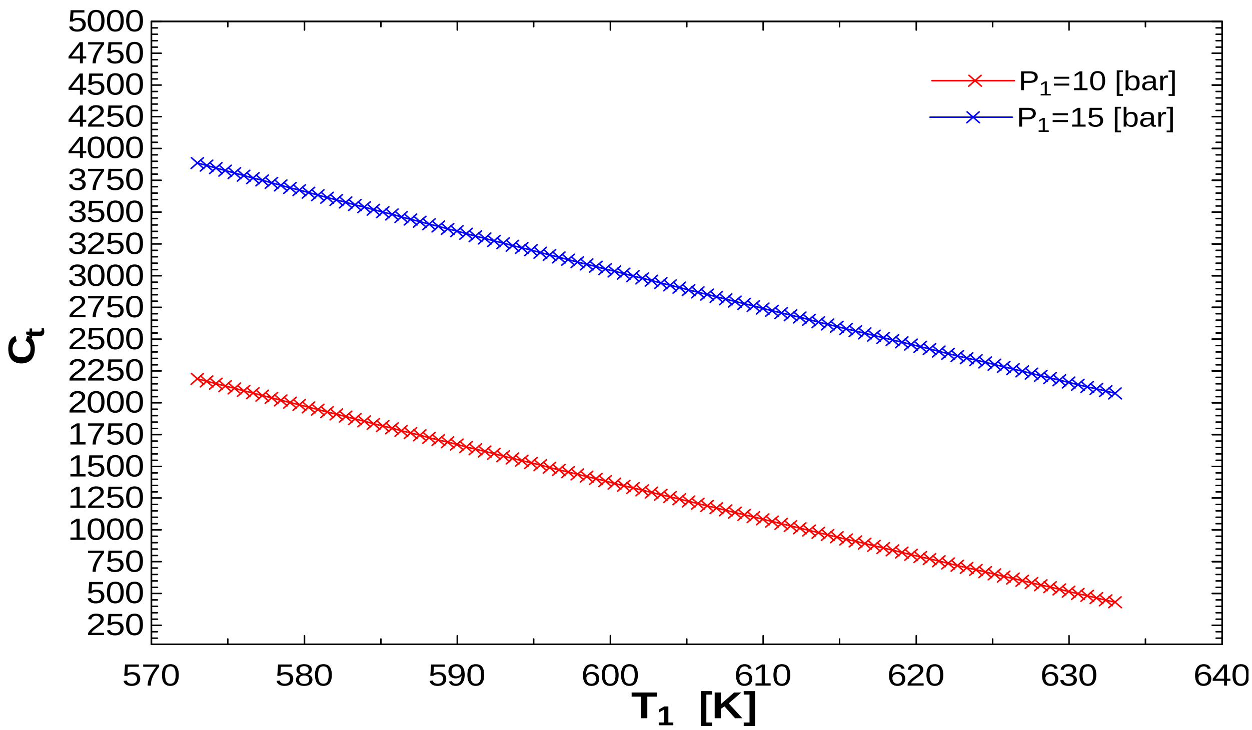


Fig. 5.24 Loss (INR) at Turbine due to exergy destruction variation with T_1 at different boiler pressure.

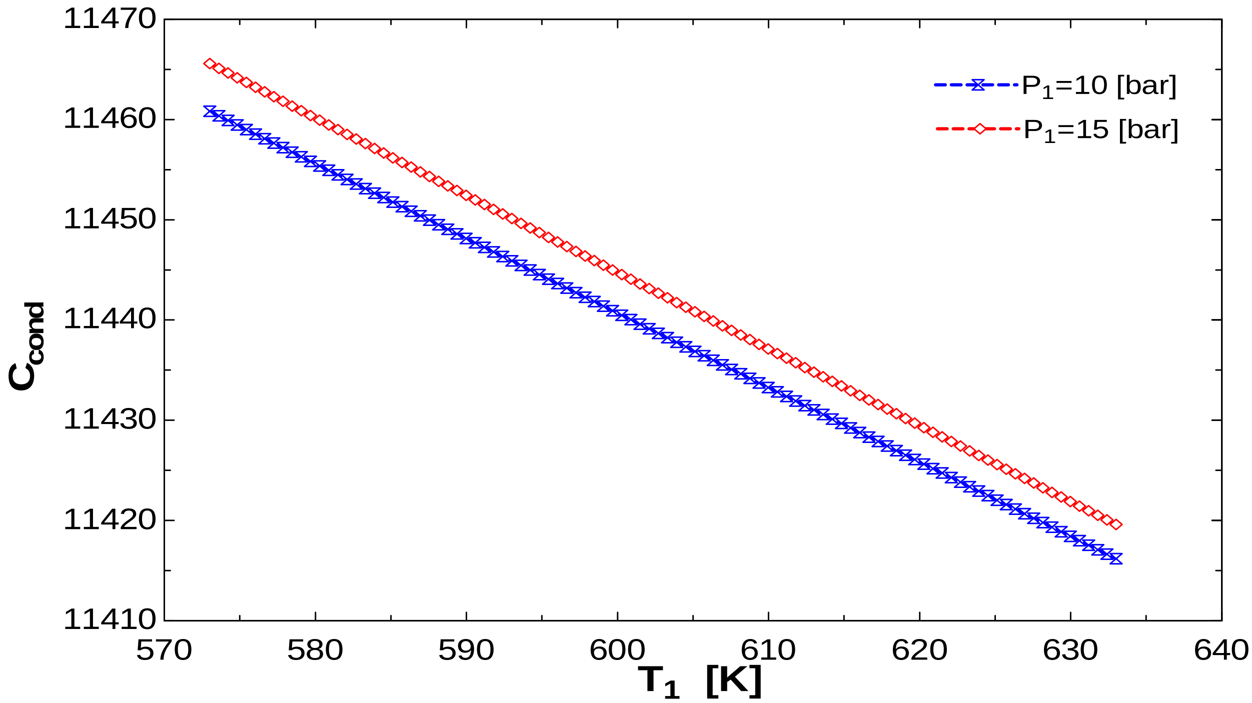


Fig. 5.25 Loss (INR) at Condenser due to exergy destruction variation with T_1 at different boiler pressure.

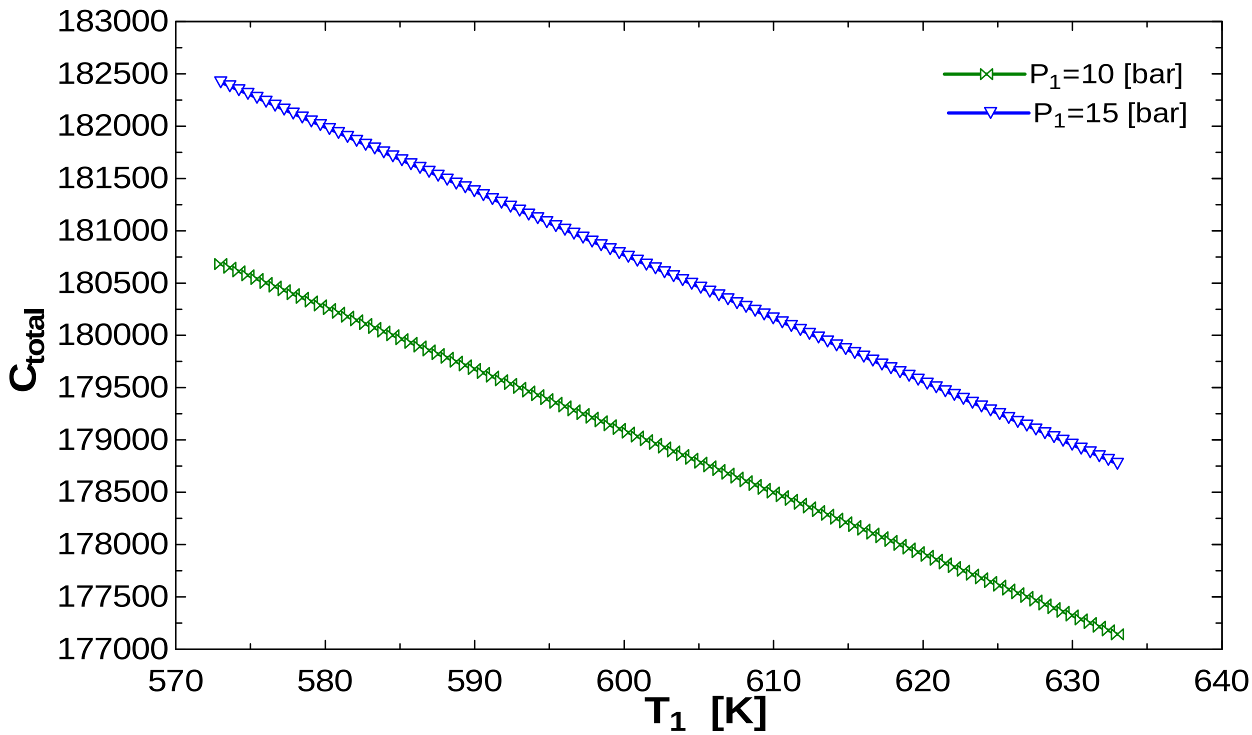


Fig. 5.26 Total loss (INR) due to exergy destruction variation with T_1 at different boiler pressure.

CHAPTER-6

CONCLUSION AND FUTURE WORK

6.1 Conclusion: The exergy analysis is important tool to find the actual irreversibilities for different components of any cycle and performance based on exergy analysis gives the real evaluation of the system**.**

In this work energy analysis and exergy analysis of coal fired steam power plant of 2000 KW at 100% load shows that the exergy destruction rate is maximum for the boiler furnace and minimum for the turbine. The exergy and energy calculated for each system in the cycle. All the relevant equation are written for the performance of calculation of different parameter that help in determining the exergetic efficiency of the components. It is found that the least value of exergetic efficiency occurs for condenser amongst all the relevant component. Thus the maximum loss occurs in the condenser and it is due to phase conversion process of the working fluid. Also major loss of exergy occurs in the condenser if our concentration is on the working cycle. Different figure are analyzed to see the variation of the exergetic efficiency, first law efficiency and exergy destruction rate with respect to steam temperature & pressure and air fuel ratio. But the transmission losses are not considered in the present work. In addition primary and secondary air temperature and pressure variation also analyzed and it is found that increasing the air inlet pressure result As the A/F ratio increases, exergy of the fuel decreases when the pressure of the primary & secondary air inducted through air preheater less than the 2bar (P_a). But as the pressure value increases above 2bar, exergy value starts increasing. Thus by increasing the inlet pressure of the air from air preheater exergy of the fuel can be achieved higher at lower value of the A/F ratio. In addition with exergy analysis, economic analysis also done in the present work. The effect of various parameters on plant economics has been observed and the following conclusions have been achieved. The total operating cost, net present value of plant, revenue earned, costs of fuel, maintenance, labour, insurance and pumping with escalation rates observed to be higher than the case without escalation. Also energy is quantified as cost and this cost is associated with the exergy loss or exergy destruction. Thus this cost shows a loss value which means exergy loss shows a monetary loss and can be minimized using the minimization in the exergy destruction.

6.2 Future work: As the exergy analysis and cost analysis are the complementary to each other, the designing of the component of the plant have been done in such a manner that results in minimum loss of exergy which means minimum loss in cost value. But to design a component for minimum exergy destruction incurs high initial cost. Thus a balance between the initial cost of component and value of cost lost due to exergy destruction can be done. Which results in the minimum fixed cost of the plant during the life of the plant.

CHAPTER-7

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