A DISSERTATION ON

SECOND LAW ANALYSIS OF SUPER CRITICAL POWER PLANT USING ENTROPY GENERATION METHOD

Submitted in partial fulfillment of the requirement

for the award of the degree of

Master of Technology

In

Thermal Engineering

Submitted By

PRASHANT BANSAL

Roll No.2K14/THE/24

Under the guidance of

DR. K. MANJUNATH

Assistant Professor

Mechanical Engineering Department

DEPARTMENT OF MECHANICAL, PRODUCTION AND AUTOMOBILE ENGINEERING

DELHI TECHNOLOGICAL UNIVERSITY,DELHI

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DECLARATION

I hereby declare that the work, which is being presented in this dissertation, entitled "SECOND LAW ANALYSIS OF SUPER CRITICAL POWER PLANT USING ENTROPY GENERATION METHOD" towards the partial fulfillment of the requirements for the award of the degree of Master of Engineering with specialization in Thermal Engineering, from Delhi technological University Delhi, is an authentic record of my own work carried out under the supervision of PROF. R.S MISHRA & DR. K.MANJUNATH Assistant Professor, Department Mechanical Engineering, at Delhi Technological University, Delhi.

The matter embodied in this dissertation report has not been submitted by me for the award of any other degree.

PRASHANT BANSAL

2K14/THE/24

Place: Delhi

Date:

This is to certify that the above statement made by the candidate is correct to the best of my knowledge.

DR. K. MANJUNATH

Assistant Professor

Department of Mechanical Engineering

Delhi Technological University

Delhi-110042

CERTIFICATE

It is certified that PRASHANT BANSAL, Roll no. 2K14/THE/24, student of M.Tech. Thermal Engineering, Delhi Technological University, has submitted the dissertation titled "SECOND LAW ANALYSIS OF SUPER CRITICAL POWER PLANT USING ENTROPY GENERATION METHOD" under my guidance towards the partial fulfillment of the requirements for the award of the degree of Master of Technology.

His work is found to be satisfactory and his discipline impeccable during the course of the project. His enthusiasm, attitude towards the project is appreciated.

I wish him success in all his endeavors.

DR. K. MANJUNATH

ASSISTANT PROFESSOR

Department of Mechanical Engineering
DELHI TECHNOLOGICAL UNIVERSITY

<u>ACKNOWLEDGMENT</u>

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.

I take great pride in expressing my unfeigned appreciation and gratitude to my learned mentors **DR. K. MANJUNATH,** and Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering), for their invaluable inspiration, guidance and continuous encouragement throughout this project work. Their critics and suggestions on my work have always guided me towards perfection. This work is simply the reflection of their thoughts, ideas, concepts and above all his efforts. Working under their guidance has been a privilege and an excellent learning experience that I will cherish for a long time.

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PRASHANT BANSAL
2K14/THE/24

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CHAPTER 1 : INTRODUCTION

1.1 ELECTRICITY SECTOR OF INDIA

A power plant is defined as the assembly of equipment that generates a flow of mechanical or electrical energy (Raja et al., 2006). The equipment used is known as the generator. Power plants are generally classified into two types: conventional power plants and non-conventional power plants, and they are classified based on the electricity generation devices and fuel type (Kitto, 1996).Power plants classified regarding the electricity generation devices such as turbines are called conventional power plants. Examples of conventional sources of energy include coal, natural gas, petroleum, and water power. The device that drives electricity generation determines the kind of power plant. For instance, steam turbine plants use the dynamic pressure generated by expanding steam to run the blades of a turbine. Some other kinds of the conventional power plants are: gas turbine plants, combined cycle plants, internal combustion plants, pulverized coal-fired power plants, circulating fluidized bed power plants, pressurized fluidized bed power plants, integrated gasification cycle power plants, hydro-electric power plants, nuclear power plants, diesel power plants, steam turbines, and steam engines (Michael et al., 2011). Among the above different kinds of power plants, steam turbines, steam engines, diesel power plants, and nuclear power plants are categorized as the thermal power plants because they convert heat into the electric energy. Power Plants that are classified based on fuel type are called non-conventional power plants. Some of the fuels used are biomass, solar, biogas, wind, tidal, and geothermal. Some examples of the power plants are thermo-electric generator, fuel cell power plants, photovoltaic solar cell power systems, fusion reactors, geothermal energy, plants, wind energy power systems, tidal wave plants, and biogas and biomass energy power system.

In india electricity sector had an installed capacity of 303 GW as of 30 Jun2016. Renewable plants constituted 28% of total installed capacity and Non-Renewable Power Plants constituted the remaining 72%. The total electricity generated by utilities is 1,106 TWh (1,106,000 GWh) and 166

TWh by captive power plants till date. Gross electricity generation includes auxiliary power consumption of power generation plants. India became the world's third largest producer of electricity in the year 2013 with 4.8% global share in electricity generation surpassing Japan and Russia

Sector	Thermal (MW)				Nuclear (MW)	Renewable (MW)			Total (MW)
	Coal	Gas	Diesel	Sub-Total Thermal		Hydel	Other Renewable	Sub-Total Renewable	
Central	51,390.00	7,555.33	0.00	55,649.73	5,780.00	11,571.43	0.00	11,571.43	76,296.76
State	64,130.50	7,210.70	363.93	71,705.13	0.00	28,092.00	1,963.81	30,055.81	101,760.94
Private	70,722.38	9,742.60	554.96	81,019.94	0.00	3,120.00	40,885.57	44,005.57	125,025.51
All India	186,242.88	24,508.63	918.89	211,670.40	5,780.00	42,783.43	42,849.38	85,632.81	303,083.21
Percentage	**61.44**	**8.08**	**0.30**	**69.83**	**1.90**	**14.11**	**14.13**	**28.25**	**100**

TABLE 1.1

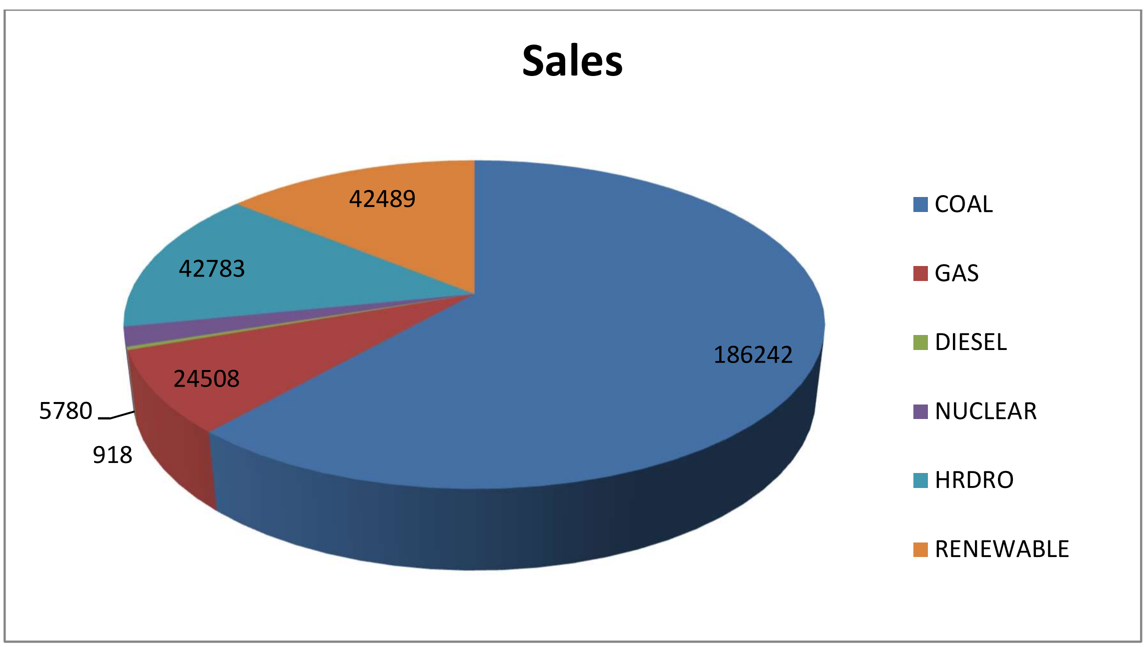


FIG 1.1

1.2 THEORY

1.2.1 DEFINITION

A thermal power station is a power plant in which the prime mover is steam driven. Water is heated , turns into steam and `rotates a steam turbine which drives an electrical generator. After it passes through turbine , the steam is condensed in a condenser and recycled to where it was heated; this is known as a Rankine cycle.

1.2.2 COMPONENTS

1) Feed water heater

2) Boiler

3) Turbine

4) Condenser

A)FEED WATER HEATER:

In the case of a conventional steam-electric power plant utilizing a drum boiler, the surface condenser removes the latent heat of vaporization from the steam as it changes states from vapor to liquid. The heat content (btu) in the steam is referred to as Enthalpy. The condensate pump then pumps the condensate water through a feed water heater. The feed water heating equipment then raises the temperature of the water by utilizing extraction steam from various stages of the turbine.

Preheating the feed water reduces the irreversibility's involved in steam generation and therefore improves the thermodynamic efficiency of the system. This reduces plant operating costs and also helps to avoid thermal shock to the boiler metal when the feed water is introduced back into the steam cycle.

B)BOILER:

Once this water is again inside the boiler or steam generator, the process of adding the latent heat of vaporization or Enthalpy is underway. The boiler transfers energy to the water by the chemical reaction of burning some type of fuel. The water enters the boiler through a section in the convection pass called the economizer. From the economizer it passes to the steam drum. Once the water enters the steam drum it goes down the down comers to the lower inlet water wall headers. From the inlet headers the water rises through the water walls and is eventually turned into steam due to the heat being generated by the burners located on the front and rear water walls.

C)TURBINE:

The turbine generator consists of a series of steam turbines interconnected to each other and a generator on a common shaft. There is a high pressure turbine at one end, followed by an intermediate pressure turbine, two low pressure turbines, and the generator. As steam moves through the system and loses pressure and thermal energy it expands in volume, requiring increasing diameter and longer blades at each succeeding stage to extract the remaining energy.

D)CONDENSER:

The steam which was used to turn the turbine is exhausted into the condenser. The steam is therefore condensed as it comes in contact with the cool tubes full of circulating water. This condensed steam is withdrawn from the bottom of the surface condenser. The condensed steam is now water, commonly referred to as condensate water.

SCHEMATIC DIAGRAM

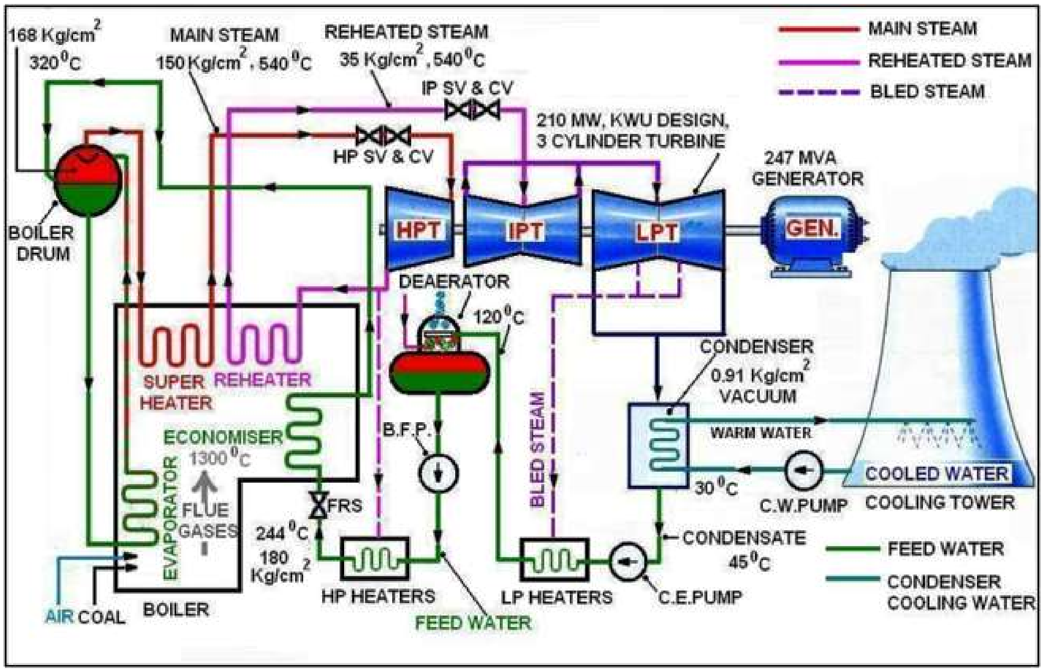


FIG 1.2

1.3 WORKING CYCLE

1.3.1 RANKINE CYCLE:

The rankine cycle is a cycle that converts heat into work. The heat is supplied externally to a closed loop, which usually uses water. There are four processes in the rankine cycle. These states are identified by numbers in diagram given below.

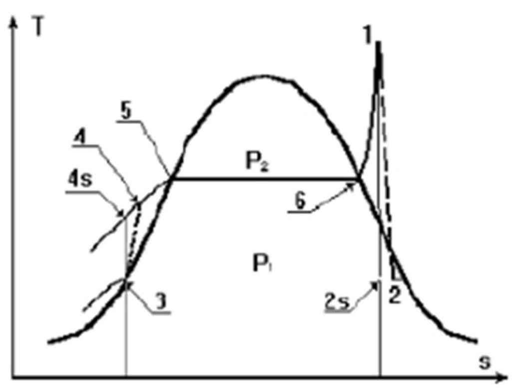


FIG. 1.3

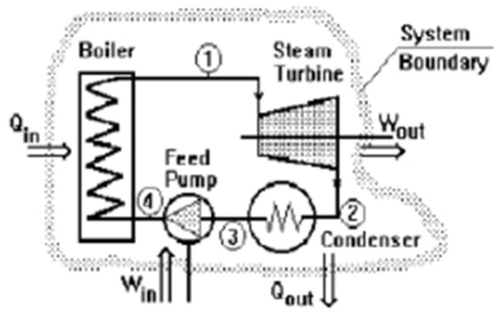


FIG. 1.4

Process 3-4 : The working fluid is pumped from low to high pressure. As the fluid is a liquid at this stage the pump requires little input energy.

Process 4-1: The high pressure liquid enters a boiler where it is heated at constant pressure by external heat sources to become a dry saturated vapor.

Process 1-2: The dry saturated vapor expands through a turbine, generating power. This decrease the temperature and pressure of the vapor, and some condensation may occur.

Process 2-3: The wet vapor then enters a condenser where it is condensed at a temperature to become a saturated liquid.

According to the T-s diagram shown in Figure, 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:

$\eta = W/Q1$

In the analysis of a rankine 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 , 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,

$m_s|q_c| = m_c c_{water} (T_{out} – T_{in})$ [kW]

Where c_{water} 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,

$T_m = (T_{out} + T_{in}) / 2$,

And the condensing temperature T_3 by the following equation;

$m_s|q_c| = UA (T_3 – T_m)$ [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:

$q = 0 = h_2 – h_1 + w_t$ [kJ/kg]

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

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

$w_t = h_1 – h_2$ [kJ/kg]

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

Turbine power = $m_s w_t = m_s (h_1 – h_2)$ [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 feedwater.

From the steady-flow First Law of

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

Work, $w_p = h_3 - h_4$ [kJ/kg]

And,

Pump power = $m_s * w_p = m_s (h_3 - h_4)$ [kW]

The pump work has a negative value as $h_4 > h_3$.

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, w_t / w_p, 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 cycleenthalpies 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.

1.4 SUPERCRITICAL

Critical is a thermodynamic expression which describes the sate of a fluid beyond which there is no clear distinction between the liquid and gaseous phase.

The critical pressure & temperature for water are Pressure -225.56 kg/cm2 Temperature - 374.15ºC

A boiler which operates at a pressure above critical point is called 'supercritical boiler'

It is a point where boiling water and dry saturated line meet so that associated latent heat is zero

1.4.1 PERCENTAGE OF SUPERCRITICAL POWER PLANTS

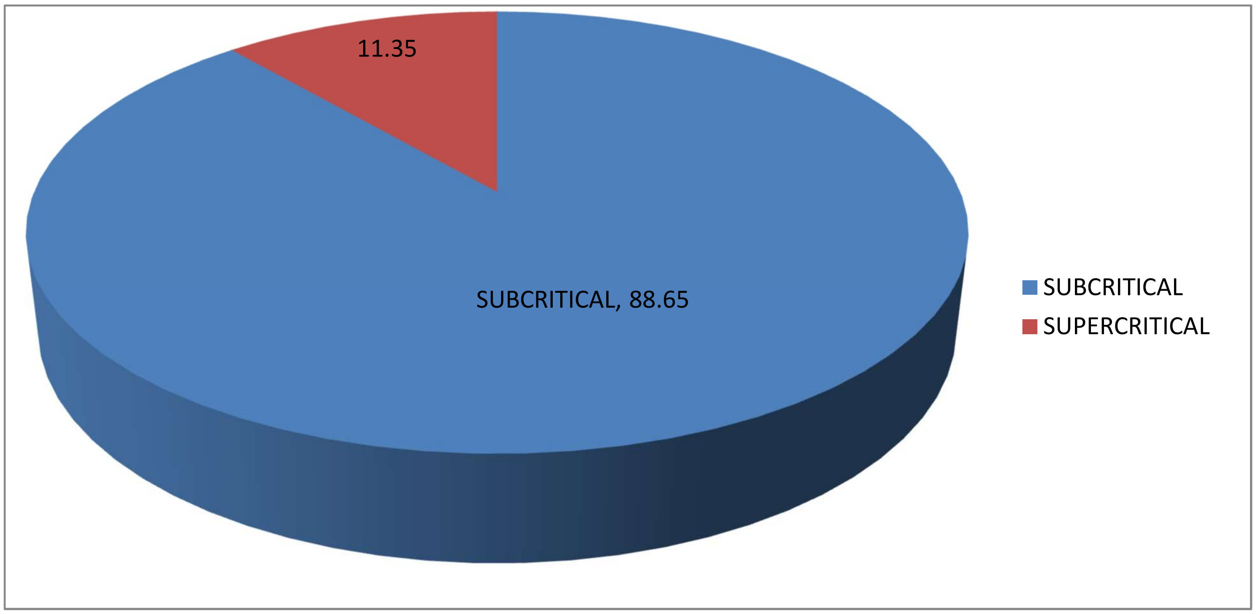


FIG. 1.5

1.4.2 NEED OF SUPER CRITICAL POWER PLANT

- Reduces emission per KWh of electricity generated.
- Rises 1% efficiency & reduces the CO_2 emission by 2-3%.
- It is most Economical way to enhance efficiency

- Fuel cost saving: Economical.
- Operating Flexibility.
- Reduced the Boiler size per MW.
- Reduced Start-Up Time.

1.4.3 How sub critical to ultra super critical march towards higher efficiency with application of higher grades of materials

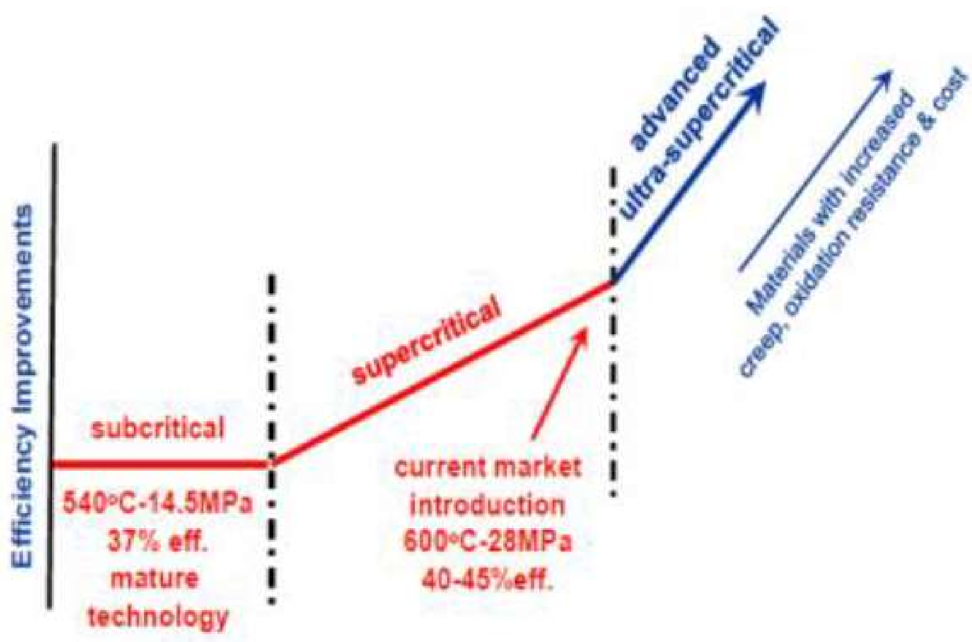


FIG. 1.6

1.4.4 SUBCRITICAL VS SUPERCRITICAL BOILER

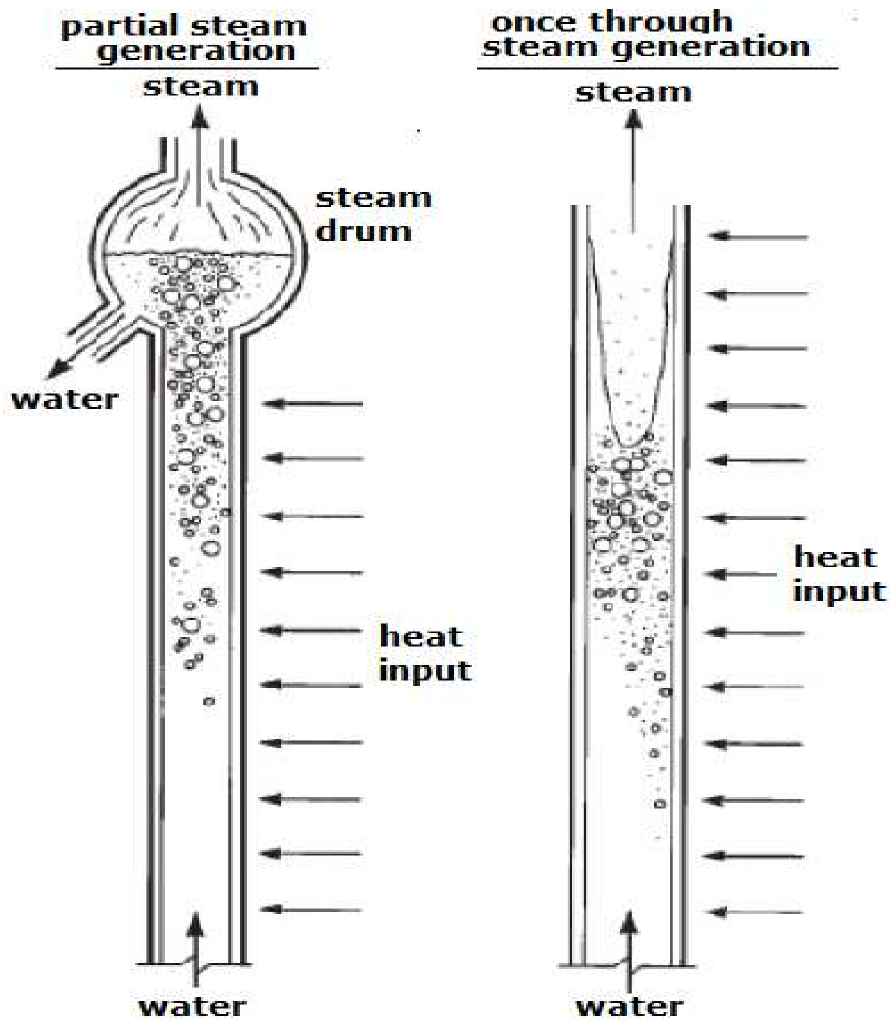


FIG.1.7

1.4.5 BOILER FOR SUPERCRITICAL ONCE THROUGH POWER PLANT

1. Once through Boiler technology, which originated in Europe, has evolved into the most effective application for Supper Critical Steam condition.

2. There are no operational limitations due to once through boilers compared through to drum type boilers.

3. In fact once-through boilers are better suited to frequent load variations than drum type boilers, since the drum is a component with a high wall thickness, requiring controlled heating. This limits the load change rate to 3% per minute, while once-through boilers can step-up the load by 5% per minute.

4. This makes once-through boilers more suitable for fast startup as well as for transient conditions.

1.4.6)CHALLENGES FOR ADOPTION OF SUPER CRITICAL TECHNOLOGY

1. Up to an operating pressure of around 190Kg Bar in the evaporator part of the boiler, the cycle is Sub-Critical. In this case a drum-type boiler is used because the steam needs to be separated from water in the drum of the boiler before it is superheated and led into the turbine.

2. Above an operating pressure of 220Kg Bar in the evaporator part of the Boiler, the cycle is Supercritical. The cycle medium is a single phase fluid with homogeneous properties and there is no need to separate steam from water in a drum.

3. Thus, the drum of the drum-type boiler which is very heavy and located on the top of the boiler can be eliminated.

4. Once-through boilers are therefore used in supercritical cycles.

5. Advanced Steel types must be used for components such as the boiler and the live steam and hot reheat steam piping that are in direct contact with steam under elevated conditions.

1.5 SECOND LAW ANALYSIS (EXERGY ANALYSIS)

Exergy is the tool, which indicates how far the system departs from equilibrium state. The concept of exergy was put forward by Gibbs in 1878. It was further developed by Rant in 1957.

1.5.1 QUALITY OF ENERGY

Quantative evaluation of energy in a cycle or in a process can be done using the first law of thermodynamics. The direction of flow of heat or work is known from the second law of thermodynamics. However, it is equally important to assign the quality to the energy. Energy can be broadly classified into high grade and low grade energy. High grade form of energy are highly organised in nature and conversion of such energy to some other high grade form (W→W) is not dictated by the second law of thermodynamics. Conversion of high grade energy to low grade energy is not desirable. However, there may be some conversion to low grade energy as work is converted into other useful form. This is because of dissipation of heat due to friction (example: mechanical work → Electricity, some losses are there due to the friction in bearing of machineries). Thus both the first and the second law of thermodynamics are to be considered for analysis. Low grade energy such as heat due to combustion, fission, fusion reactions as well as internal energies are highly random in nature. Conversion of such form of energy into high grade energy (Q→W) is of interest. This is due to the high quality of organised form of energy obtained from low quality energy. Second law of thermodynamics dictates that conversion of 100% heat into work is never possible. That part of low grade energy which is available for conversion is termed as available energy, availability or exergy. The part, which according to the second law of thermodynamics, must be rejected is known as unavailable energy. Exergy analysis helps in finding the following:

It can be used to determine the type, location and magnitude of energy losses in a system

It can be used to find means to reduce losses to make the energy system more efficient

At this point, it is worth mentioning that the environment plays an important role in evaluating the exergy (composite property).

1.5.2 IMPORTANCE OF EXERGY ANALYSIS

Carnot first detailed the concept of exergy analysis in the year 1824 followed by Clasius in the year 1865. The first law of thermodynamics (first law of analysis) is applied to a system or process to determine the efficiency, whereas the second law of analysis (exergy analysis) is based on the concept of irreversibility and entropy production. Researchers are constantly applying the concept of the first law of analysis to calculate the energy losses using the enthalpy balance over the coal-fired power. In recent years, the concept of exergy analysis has gained importance over the first law of analysis as the first law of thermodynamic analysis fails to produce sufficient results to study the performance of a power plant. The objective of exergy analysis is to identify the locations of exergy losses and to study the quality of the power plant. Exergy analysis stands as a tool to locate the imperfections inside the process or system, which first law of analysis fails to do. Exergy analysis not only locates the irreversibility inside the system, but it also helps in assessing the efficiency of the individual component. Findings from exergy analysis can help process practitioners refine and develop the optimal coal-fired power plant.

Let us take an example. In case of coal fired power plant, the first law indicates that the condenser greatly effects the power plant efficiency as large amount of heat is transferred to the cooling water without providing any clue on the real usefulness of this relatively low temperature fluid. Also, energy balances do not provide information about the internal losses such as throttling valve and heat exchanger. Second law or exergy balance, however indicates that there is hardly 1% exergy loss in the condenser with more than 60% in the boiler. The contribution in the boiler exergy loss accounts for irreversibilities associated with combustion and finite temperature differences. Hence, analysis of exergy plays a deterministic role in identification of processes and rectifying the components

CHAPTER 2

2.1 LITERATURE REVIEW

The electricity sector in India supplies the world's 6th largest energy consumer, accounting for 3.4% of global energy consumption by more than 17% of the global population. Due to the fast paced growth of Indian economy there has been an average increase of 3.6% in the energy demand per annum over the last 30 years.

mukesh gupta & raj kumar [1] puts forward the exergy based analysis of a power plant as the way to approach the analysis of a power plant. The study points out that the boiler, combustor and turbine are the critical components where maximum exergy losses occur. The first law analysis shows major energy loss has been found to occur in condenser. The second law (Exergy) analysis shows that combustion chamber in both steam and gas turbine thermal power plants are main source of Irreversibility. The Irreversibility in condenser is insignificant, because in the condenser the low quality energy is lost. An Exergy method of optimization gives logical solution improving the power production opportunities in thermal power plants.

I.Satyanarayana & Dr.A.V.S.S.K.S. Gupta[2] The energy and exergy analyzes of the cycle has been performed pressure range between 200bar to 425 bar and temperature range are 5000C-8000C. First law analysis and second law analysis has carried out throw with and withot reheat. The irreversibility and fractional exergy loss are determined for the cycle with and without reheat. It is found that the cycle efficiency is high in reheat than the nonreheat supercritical cycle. It is also conclude that exergy efficiency is high in reheat than non-reheat supercritical cycle. It is found that nearly 20-25% irreversibility is reduced by using single reheat in the boiler, where as it is 12-15% in the turbine than the without reheating. Fractional exergy losses of all the components in the cycle is determined and compared with and without supercritical cycle.

M. Pandey[3] concluded that The energy efficiency of the cycle increases as a result of using regeneration, open FWH and closed FWH. From the exergy analysis it is found that the losses due to irreversibility were maximum in the boiler than in the turbine followed by the condenser. Further it was seen that the fractional irreversibility in the boiler increases with turbine inlet temperature whereas it decrease with increase in boiler pressure. In the present work a simple reheat-regenerative rankine cycle was considered for calculation of energy efficiency, exergy efficiency and the irreversible losses.

S C KAUSHIK[4] perform an energetic and exergetic analysis on a 660 MWe coal fired supercritical thermal power plant at 100%, 80% and 60% of normal continuous rating (NCR) conditions under constant pressure as well as pure sliding pressure operation and to highlight the benefits of the latter over the former. The energetic input, energetic output, exergetic input, exergetic output, energetic and exergetic efficiencies of various components of the supercritical thermal power plant are estimated at 660 MWe, 528 MWe and 396 MWe load under both constant pressure as well as pure sliding pressure operation. Also the energy losses and exergy destruction in various components of a power plant i.e. Boiler, high pressure turbine (HPT), intermediate pressure turbine (IPT) , low pressure turbine (LPT), condenser, gland steam coolers, condensate extraction pumps, low pressure heaters (LPH), drip pumps (DP), deaerator (D), boiler feed pump (BFP) and high pressure heaters (HPH) have been calculated. The results have shown that the boiler has the maximum rate of exergy destruction than any other component in the power plant. After the boiler, turbine has the maximum rate of exergy destruction than any other component of the power plant. The study reveals that there is a significant reduction in the rate of exergy destruction at part load conditions for the turbine in case of sliding pressure operation in comparison to constant pressure operation.

S C KAUSHIK & S K TYAGI[5] carried out the energetic and exergetic analysis of natural gas fired combined cycle power plant, Linear Fresnel Reflecting Solar Concentrator and also solar aided natural gas fired combined cycle power plant. The exergetic analysis shows that combustion chamber subsystem followed by heat recovery steam generator (HRSG) is main source of exergy loss in a natural gas fired combined cycle power plant. The exergetic power loss in the condenser is less. In the analysis linear Fresnel reflecting solar concentrator (LFRSC) by increasing focal distance and decreasing width linear Fresnel reflector, total reflector area come close to actual reflector area. By the increasing of normal incidence beam solar radiation (Ib) both energetic and exergetic efficiencies increases. Increasing inlet temperature and decreasing the mass flow rate of thermic fluid exergetic efficiency is increased.

SANDHYA HASTI & ADISORN AROONWILAS[6] performed exergy analysis for ultra super-critical power plant. The analysis was carried out by means of process simulation using a computer model developed in Microsoft Excel .The model was based on the concepts of coal combustion, energy balances, enthalpy balances, entropy changes and heat transfer of the steam power cycle. After development, the validated model was used to simulate the hypothetical power plant combusting lignite coal with the net output of 422 MW. The exergy loss indicates that the highest concentration of losses appears to be the furnace followed by the turbine.

M.K. Gupta , S.C. Kaushik [7] carried out energy and exergy analysis for the different components of a proposed conceptual direct steam generation (DSG) solar–thermal power plant (STPP). It has been found that the maximum energy loss is in the condenser followed by solar collector field.The maximum exergy loss is in the solar collector field while in other plant components it is small. The possibilities to further improve the plant efficiency are identified and exploited. For minimum exergy loss in receiver the inlet temperature of water to the receiver, which is governed by the number of feed water heaters (FWHs), bleed pressure and mass fraction of bleed steam, must be optimum.

V. Siva Reddy , S.C. Kaushik , S.K. Tyagi[8] carried out energetic and exergetic analysis for natural gas fired combined cycle power plant, Linear Fresnel Reflecting Solar Concentrator and also solar aided natural gas fired combined cycle power plant. The exergetic analysis shows that combustion chamber subsystem followed by heat recovery steam generator (HRSG) is main source of exergy loss in a natural gas fired combined cycle power plant. It is observed that the utilization of solar energy for feed water heating and low pressure steam generation is more effective based on exergetic analysis rather than energetic analysis.

Ligang Wang 1, Yongping Yang 1, Tatiana Morosuk 2 and George Tsatsaronis[9] in this paper both conventional and an advanced exergy analysis is performed on supercritical coal-fired power plant. The results show that the ratio of exogenous exergy destruction differs quite a lot from component to component. In general, almost 90% of the total exergy destruction within turbines comes from their endogenous parts, while that of feedwater preheaters contributes more or less 70% to their total exergy destruction. Moreover, the boiler subsystem is proven to have a large amount of exergy destruction caused by the irreversibilities within the remaining components of the overall system. It is also found that the boiler subsystem still has the largest avoidable exergy destruction.

S.K. Soma, A. Dattab[10] makes a comprehensive review pertaining to

fundamental studies on thermodynamic irreversibility and exergy analysis in the processes of combustion of gaseous, liquid and solid fuels. The need for such investigations in the context of combustion processes in practice is first stressed upon and then the various approaches of exergy analysis and the results arrived at by different research workers in the field have been discussed. It has been recognized that, in almost all situations, the major source of irreversibility's is the internal thermal energy exchange associated with high temperature gradients caused by heat release in combustion reactions. The primary way of keeping the exergy destruction in a combustion process within a reasonable limit is to reduce the irreversibility in heat conduction through proper control of physical processes and chemical reactions resulting in a high value of flame temperature but lower values of temperature gradients within the system.

V. Raghavan a, G. Gogos a, V. Babub, T. Sundararajan b[11] has performed entropy generation during the quasi-steady combustion of spherical liquid fuel particles .The effects of free stream velocity, particle diameter, ambient temperature and gravity, on the entropy generation rate, have been discussed in detail. In the range of subcritical freestream velocity, where an envelope flame is present, the entropy generation rate presents a minimum value. At a critical velocity, where the flame transition occurs, the entropy generation rate reaches a maximum value. Flame transition significantly affects the entropy generation rate, which suffers a sharp decrease in its value after the transition.

Meeta Sharma , Onkar Singh [12] has performed exergy analysis of a dual pressure (DP) heat recovery steam generator (HRSG) having steam generation at high pressure (HP) and low pressure (LP) in the range of 50–70 bar and 2–6 bar respectively in the gas/steam combined cycle power plant for varying dead states. The in-operation plant data for this study are taken from a gas/steam combined cycle power plant at Auraiya (U.P.), India. Results have been obtained for exergy loss and exergy efficiency with varying dead state temperatures for different HP and LP steam generation states in different sections of HRSG. The exergy analysis for chosen conditions/parameters helps in locating the particular sections of the HRSG having maximum exergy loss. It is found that at varying steam generation pressures the HP and LP super heater sections and at higher dead state temperatures the HP evaporator are found to act as major source of irreversibilities.

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.

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.

2.2 Exergy Analysis

The thermodynamic analysis of any power plant includes the material, energy, entropy, and exergy balances . It is important to determine the amount of work potential that can be achieved from the system. The maximum work potential obtained from the system at any given reference temperature and pressure is often referred to “useful work or exergy”. As exergy deals with the amount of useful work available in the system with respect to the reference point, exergy is also termed as “availability or available energy” The amount of useful work produced depends on the conditions of the system and the immediate surroundings outside the system. At steady state, the exergy balance equation for the control volume system given in Figure 2.3 can be written as

$$\sum_{in} \dot{m}\,\Psi + \dot{Q} - \sum_{Out} \dot{m}\Psi - \dot{W} = 0$$

where Ψ, $\dot{Q}$, and $\dot{W}$ represent specific exergy, heat transfer rate, and work rate or power, respectively. Entropy can be defined as a state variable whose change is defined for a reversible process at a given temperature. The entropy change relation for the control volume system depends on the inlet entropy and outlet entropy difference.

$\sum_{in}$ where S_{in}, the outlet,

$$\dot{S}_{in} + \sum_{in}\frac{\dot{Q}_{in}}{T} + \dot{S}_{gen} = \sum_{out}\dot{S}_{out} + \sum_{out}\frac{\dot{Q}_{out}}{T}$$

$\dot{S}_{in}$, $\dot{S}_{out}$, $\dot{S}_{gen}$, $\dot{Q}_{in}$, $\dot{Q}_{out}$ and T represent the entropy rate at the inlet, entropy rate at entropy rate generated inside the system, heat rate at the inlet, heat rate at the outlet, and temperature of the system. The entropy generation number can be written as

$$N_s = \frac{\dot{S}_{gen}}{C_p . \dot{m}_g}$$

where N_s is the entropy generation number, C_p is the specific heat value, and $\dot{m}_g$ is the mass flow rate of the gas. The general equation for the unit exergy can be written as

$$\Psi = (h - h_0) - T_o (s - s_o)$$

where h, s represent the specific enthalpy and specific entropy at temperature T and h_0 , s_0 and T_0 are the specific enthalpy, specific entropy, and temperature of the reference property. The destruction of exergy in a system depends on the entropy generation or changes happening within the system considered. Whenever an entropy change is generated inside the system, a part of the useful work that can be done by the system is destroyed, which indirectly leads to energy loss, and this destruction of energy is known as Exergy destruction (X_{d}).

When the entropy generated is less, then the exergy destruction rate is also less,

and when the entropy generated is greater, the destruction is greater. Thus, we can conclude that exergy destruction rate depends on entropy generation rate. The general equation of exergy destruction for the system can be expressed as

$$\dot{X}_d = T_o . \dot{S}_{gen}$$

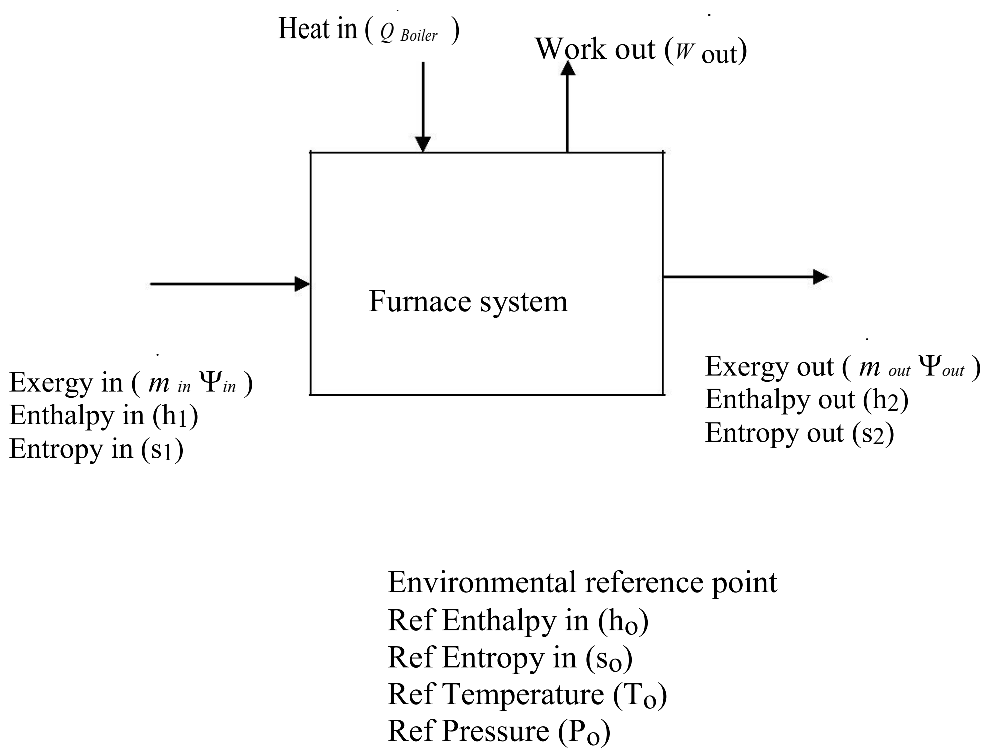


Figure 1.7 Scheme of furnace explaining the exergy balance

Exergy is developed based on the second law of thermodynamics where the analysis depends on the entropy generation of the system. The entropy generation depends on the inlet and the outlet entropy changes during the process.

$$\dot{S}_{gen} = \dot{m}_s \left(s_2 - s_1 + \frac{q_{surr}}{T_o}\right)$$

where m_s represent the flow rate of the steam entering the system and q_{surr} is the heat loss due to the surroundings. For the control volume system at any given condition, the general exergy balance equation is given by the relation below.

$$\sum_{in}\left(1 - \frac{T}{T_k}\right)\dot{Q}_k + \sum_{in}\dot{m}\dot{\Psi} = \dot{W} + \dot{X}_d + \sum_{out}\dot{m}\dot{\Psi}$$

where $\dot{Q}_k$ represent the heat transfer rate of the surroundings and T_k is the surrounding temperature.

For any process, there are certain losses like heat loss and surrounding loss. Heat loss is due to the heat and mass transfer inside the system and the losses that occur due to the surroundings is known as surrounding losses. As a result of the heat loss, there is loss in the work done.

2.3 Exergy Analysis of Integrated Furnace System

The exergy balance inside the furnace system can be written as

$$\dot{m}_a\Psi_a + \dot{m}_c\Psi_c + \dot{m}_4\Psi_4 + \dot{m}_{31}\Psi_{31} - \dot{m}_{28}\Psi_{28} - \dot{m}_{29}\Psi_{29} - \dot{m}_c.N_{fluegas}.\Psi_E = \dot{X}_{d,F}$$

where $\Psi_{28}, \Psi_4, \Psi_{29}, \Psi_{31}, \Psi_E$ represents the exergy at superheater out, feedwater in, reheater out, reheater in, and flue gas out from the economizer, respectively. The exergy balance of the heat exchanger unit is given as

$$\dot{m}_4\Psi_4 + \dot{m}_{31}\Psi_{31} - \dot{m}_{28}\Psi_{28} - \dot{m}_{29}\Psi_{29} \quad = X_{d,\,Boiler}$$

The exergetic efficiency of the furnace is given as

$$\eta_{\Psi,\,Furnace} = \frac{(\dot{m}_{28}\Psi_{28} - \dot{m}_4\Psi_4) + (\dot{m}_{29}\Psi_{28} - \dot{m}_{31}\Psi_{31})}{(\dot{m}_a\Psi_a + \dot{m}_c\Psi_c) - (\dot{m}_{c.}\,N_{fluegas}\,.\Psi_E)}$$

2.4 Exergy Analysis of Turbines and pumps

The exergy balance for the turbines and pumps can be written as

$$\sum_{i=1}^{m} (\dot{m}_{s,i}\,.\Psi_i)_{in} - \sum_{i=1}^{n.} (\dot{m}_{s,i}\,.\Psi_i)_{out} - \dot{W}_{T,\,total} = \dot{X}_{d,\,T}$$

where i represents the steam extraction stage at any point of the turbine and Ψ_i denotes the exergy of the stream i. The exergetic efficiency of the turbine is given as

$$\eta_{\Psi,Turbine} = \frac{\dot{W}_{T,\,total}}{\sum_{i=1}^{m}(\dot{m}_{s,i}\,.\Psi_i)_{in} - \sum_{i=1}^{n.}(\dot{m}_{s,i}\,.\Psi_i)_{out}}$$

The exergy balance around the condenser pump can be written as

$$\dot{m}_s\,(\Psi_{CP,\,in} - \Psi_{CP,\,Out}) + \dot{W}_{CP} = \dot{X}_{d,CP}$$

where $\Psi_{CP,\,in}, \Psi_{CP,\,out}$ represents the exergy at the condensate pump inlet and outlet, respectively. The exergetic efficiency of the condenser pump is given as

$$\eta_{\Psi,CP} = \frac{\dot{m}_s\,(\Psi_{CP,\,in} - \Psi_{CP,\,out})}{\dot{W}_{CP}}$$

The exergy balance for the boiler feedwater pump can be written as

$$\dot{m}_w\,(\Psi_{BP,\,in} - \Psi_{BP,\,out}) + \dot{W}_{BP} = \dot{X}_{d,\,BP}$$

where $\Psi_{BP,\,in}, \Psi_{BP,\,out}$ represents the exergy at the boiler feedwater pump inlet and outlet, respectively.

The exergetic efficiency for the boiler feedwater pump can be given as

$$\eta_{\Psi,BP} = \frac{\dot{m}_w (\Psi_{BP,in} - \Psi_{BP,out})}{\dot{W}_{BP}}$$

2.5 Exergy Analysis of Condenser

The exergy balance for the condenser can be written as,

$$\dot{m}_s (\Psi_{s,In} - \Psi_{s,Out}) - \dot{m}_{cw} (\Psi_{cw,In} - \Psi_{cw,Out}) = \dot{X}_{d,CND}$$

where $\Psi_{s,In}$, $\Psi_{s,out}$, $\Psi_{cw,In}$, $\Psi_{cw,Out}$ represents the exergy of steam in, steam out, cooling water in, and cooling water out, respectively. The exergetic efficiency of the condenser is given as

$$\eta_{\Psi,CND} = \frac{\dot{m}_{cw} (\Psi_{cw,in} - \Psi_{cw,Out})}{\dot{m}_s (\Psi_{s,In} - \Psi_{s,Out})}$$

CHAPTER 3

3.1 RESULT AND DISCUSSION

After the analysis of different cycles on which the super critical thermal power plant work with respect to energy, exergy. 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 destruction shows a loss, which can be quantify by analysis the system in mathematically. In the present work the analysis is done in the super critical steam power plant.

Result from Energy and Exergy analysis:

3.1.1) Effect of turbine inlet temperature T_1 at boiler pressure P_1 (bar) on-

- **Work output from the turbine:** The variation of the W_t with $T_1$is shown in Fig. 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 Fig. 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 Fig. 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 Fig. Exergy destruction decreases with increase in T_1. On increasing the value of boiler pressure the value of exergy destructionat a particular value of T_1, increases.
- **Second law efficiency of condenser:** The variation of the η_{2cond} with T_1 is shown in Fig. 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 Fig. Exergy destruction decreases with increase in T_1. On increasing the value of boiler pressure the value of exergy destructionat a particular value of T_1, increases.

- **Exergy destruction in boiler:**The variation of the I_{boiler} with T_1 is shown in Fig. Exergy destruction decreases with increase in T_1. On increasing the value of boiler pressure the value of exergy destructionat a particular value of T_1, increases
- **Exergy destruction in pump:**The variation of the I_{pump} with T_1 is shown in Fig. Exergy destruction decreases with increase in T_1. On increasing the value of boiler pressure the value of exergy destructionat 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 flowrate of cooling water increases with increase in T_1. On increasing the value of boiler pressure the value of mass flowrate of waterat a particular value of T_1, decreases.

- **Second law eff. Of cycle :** The variation of the Second law eff. Of cycle with T_1 is shown in Fig. Second law eff. Of cycle increase with increase in T_1. On increasing the value of boiler pressure the value of a Second law eff. Of cycle particular value of T_1, decreases.

- **first law eff. Of cycle :** The variation of the first law eff. Of cycle with T_1 is shown in Fig. first law eff. Of cycle increase with increase in T_1. On increasing the value of boiler pressure the value of a first law eff. Of cycle particular value of T_1, decreases.

3.1.2)Effect of mass flow rateof steam at boiler pressure P_1 (bar) on-

- **Work output from the turbine:** The variation of the W_t with M_s is shown in Fig. Work output decrease with increase in M_s. On increasing the value of boiler pressure the value of work output at a particular value of M_s, increase
- **First law efficiency of turbine:** The variation of the η_{t1} with M_s is shown in Fig. First law efficiency decrease with increase in M_s. On increasing the value of boiler pressure the value of first law efficiency at a particular value of M_s increase.
- **Second law efficiency of turbine:** The variation of the η_{2t} with M_s is shown in Fig. Second law efficiency decrease with increase in M_s On increasing the value of boiler pressure the value of second law efficiency at a particular value of M_s, increase..
- **Exergy destruction in turbine:** The variation of the I_t with M_s is shown in Fig. Exergy destruction increase with increase in M_s. On increasing the value of boiler pressure the value of exergy destructionat a particular value of M_s, decrease

- **Second law efficiency of condenser:** The variation of the η_{2cond} with M_s is shown in Fig. Second law efficiency decrease with increase in M_s. On increasing the value of boiler pressure the value of second law efficiency at a particular value of M_s , increase.
- **Exergy destruction in condenser:**The variation of the I_{cond} with M_s is shown in Fig. Exergy destruction increase with increase in M_s. On increasing the value of boiler pressure the value of exergy destructionat a particular value of M_s, decrease
- **Exergy destruction in boiler:**The variation of the I_{boiler} with M_s is shown in Fig. Exergy destruction increase with increase in M_s. On increasing the value of boiler pressure the value of exergy destructionat a particular value of M_s, decrease
- **Exergy destruction in pump:**The variation of the I_{pump} with M_s is shown in Fig. Exergy destruction increase with increase in M_s. On increasing the value of boiler pressure the value of exergy destructionat a particular value of M_s, decrease

T_1	X_1	X_2	X_3	X_4	M_S	m_{cw}	W_{NET}	$FIRST_{LAW,\eta,TOT}$	P_1	$\eta_{SECOND,LAW}$	$\eta_{CONDENSOR,SE}$	$\eta_{TURBINE,2}$
575.3	56747	19691	3404	398.4	432	15937	763.9	0.2482	230	0.9397	0.9963	0.9453
576.3	56563	19581	3390	397.1	430.3	15875	766.8	0.2489	230	0.9397	0.9963	0.9456
577.3	56381	19472	3377	395.8	428.7	15815	769.7	0.2496	230	0.9398	0.9963	0.9459
578.3	56201	19364	3364	394.6	427.1	15755	772.6	0.2503	230	0.9398	0.9963	0.9462
579.3	56022	19258	3351	393.4	425.5	15695	775.5	0.2511	230	0.9399	0.9963	0.9465
580.3	55844	19152	3338	392.1	423.9	15636	778.4	0.2518	230	0.9399	0.9963	0.9467
581.3	55669	19047	3325	390.9	422.4	15577	781.3	0.2525	230	0.94	0.9963	0.947
582.3	55494	18943	3313	389.7	420.8	15519	784.2	0.2532	230	0.9401	0.9963	0.9473
583.3	55321	18840	3300	388.5	419.3	15462	787.1	0.2539	230	0.9401	0.9963	0.9475
584.3	55150	18738	3287	387.3	417.8	15405	789.9	0.2546	230	0.9402	0.9963	0.9478
585.4	54980	18637	3275	386.2	416.2	15348	792.8	0.2553	230	0.9402	0.9963	0.9481
586.4	54811	18537	3263	385	414.7	15292	795.7	0.256	230	0.9403	0.9963	0.9483
587.4	54644	18438	3251	383.8	413.3	15236	798.5	0.2567	230	0.9403	0.9963	0.9486
588.4	54478	18339	3239	382.7	411.8	15181	801.4	0.2574	230	0.9404	0.9963	0.9488
589.4	54314	18242	3227	381.6	410.3	15127	804.3	0.2581	230	0.9404	0.9963	0.9491
590.4	54151	18145	3215	380.4	408.9	15072	807.1	0.2587	230	0.9405	0.9963	0.9493
591.4	53989	18050	3203	379.3	407.4	15019	810	0.2594	230	0.9406	0.9963	0.9496
592.4	53829	17955	3192	378.2	406	14965	812.8	0.2601	230	0.9406	0.9963	0.9498
593.4	53669	17860	3180	377.1	404.6	14912	815.7	0.2608	230	0.9407	0.9963	0.9501
594.4	53511	17767	3168	376.1	403.2	14860	818.5	0.2615	230	0.9407	0.9963	0.9503
595.5	53355	17675	3157	375	401.8	14808	821.3	0.2621	230	0.9408	0.9963	0.9506
596.5	53199	17583	3146	373.9	400.4	14756	824.2	0.2628	230	0.9408	0.9963	0.9508
597.5	53045	17492	3135	372.9	399	14705	827	0.2635	230	0.9409	0.9963	0.951
598.5	52892	17402	3124	371.8	397.7	14654	829.8	0.2642	230	0.9409	0.9963	0.9513
599.5	52740	17312	3113	370.8	396.3	14604	832.6	0.2648	230	0.941	0.9963	0.9515

TABLE 1.2

PARAMETRIC TABLE FOR DIFFERENT VALUES OBTAINED USING EES SOFTWARE AT DIFFERENT TURBINE INLET TEMP.

I_1	I_2	I_3 [kJ/kg-K]	I_4	T_1
69.65	25.61	4.251	0.4888	551
69.7	25.56	4.25	0.4891	552
69.74	25.52	4.25	0.4895	553
69.79	25.47	4.25	0.4898	554
69.84	25.42	4.249	0.4901	555.1
69.88	25.38	4.249	0.4904	556.1
69.93	25.33	4.248	0.4907	557.1
69.97	25.29	4.248	0.4911	558.1
70.02	25.24	4.248	0.4914	559.1
70.06	25.2	4.247	0.4917	560.1
70.11	25.15	4.247	0.492	561.1
70.15	25.11	4.247	0.4923	562.1
70.2	25.06	4.246	0.4927	563.1
70.24	25.02	4.246	0.493	564.1
70.29	24.98	4.246	0.4933	565.2
70.33	24.93	4.245	0.4936	566.2
70.37	24.89	4.245	0.4939	567.2
70.42	24.84	4.244	0.4943	568.2
70.46	24.8	4.244	0.4946	569.2
70.51	24.76	4.244	0.4949	570.2
70.55	24.71	4.243	0.4952	571.2
70.59	24.67	4.243	0.4955	572.2
70.64	24.63	4.243	0.4959	573.2
70.68	24.58	4.242	0.4962	574.2

TABLE 1.3PERCENTAGE OF EXERGY LOSS IN MAJOR FOUR COMPONENTS OF POWER PLANT

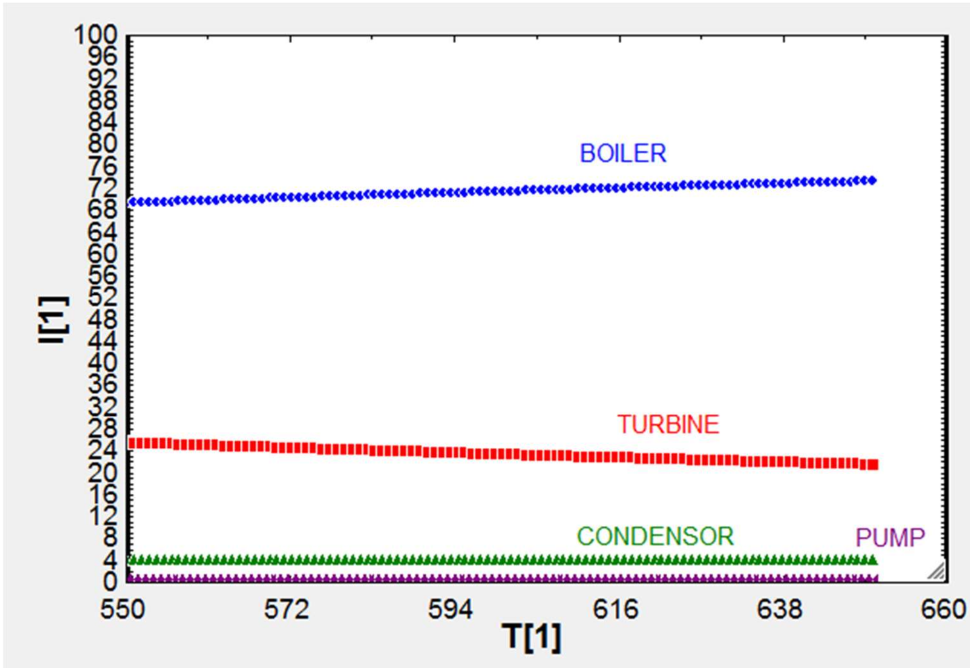


FIG1.7 EXERGY LOSS IN MAJOR FOUR COMPONENTS OF POWER PLANT VS. TURBINE INLET TEMP.

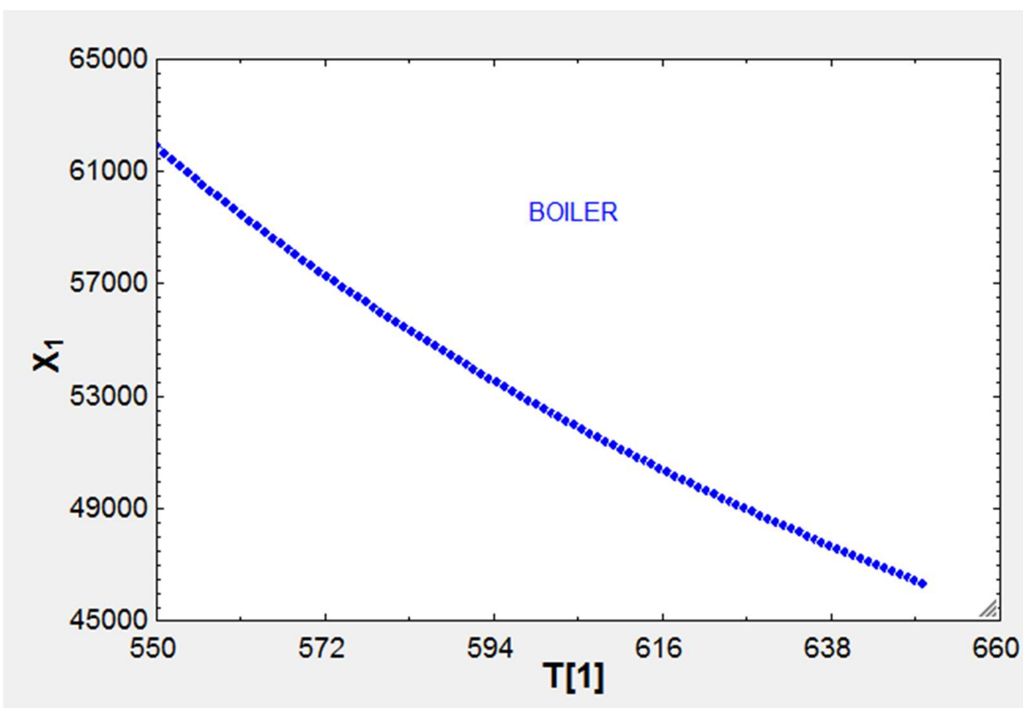


FIG 1.8 EXERGY LOSS IN BOILER VS. TURBINE INLET TEMP.

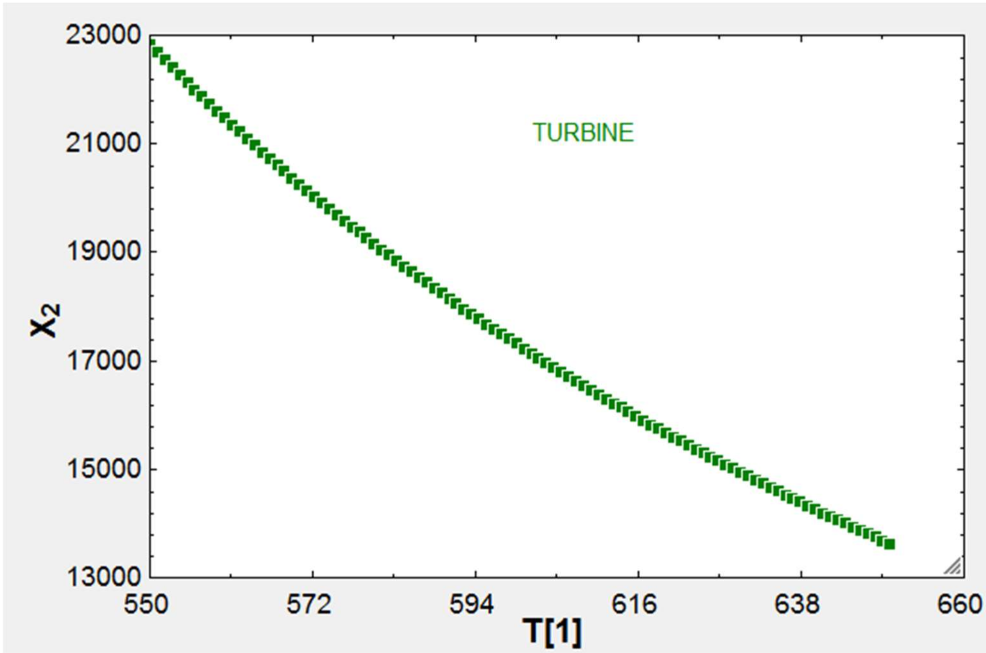


FIG1.9 EXERGY LOSS IN TURBINE VS. TURBINE INLET TEMP.

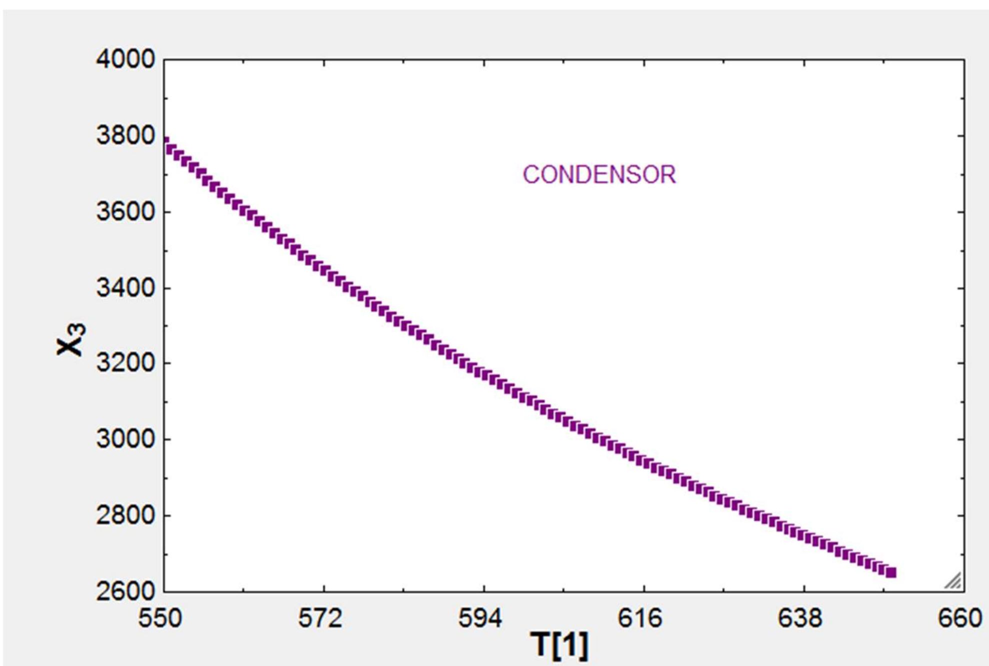


FIG 1.10 EXERGY LOSS IN CONDENSOR VS. TURBINE INLET TEMP.

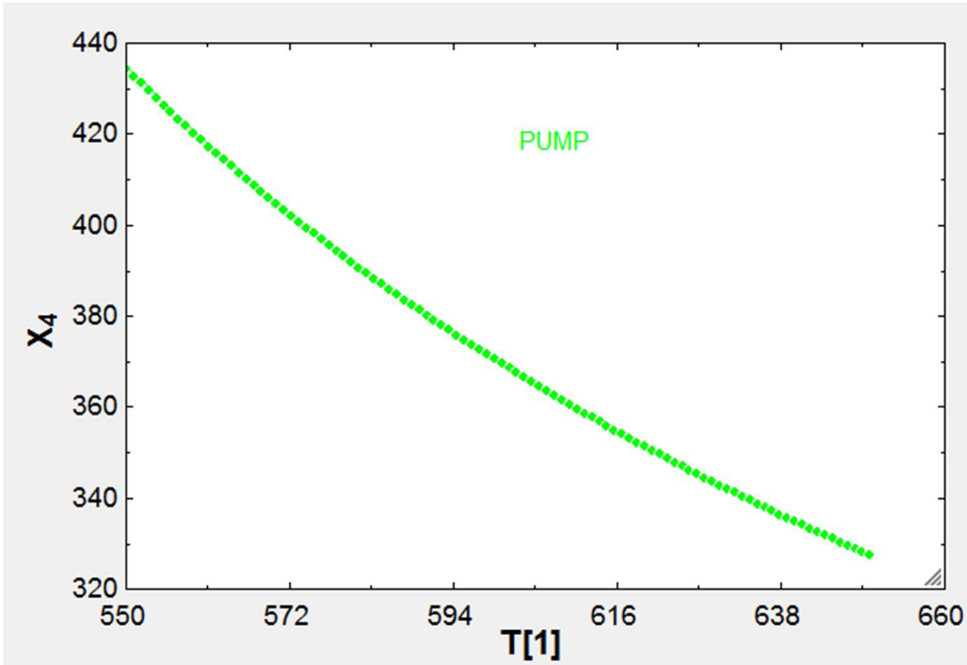


FIG 1.11 EXERGY LOSS IN PUMPVS. TURBINE INLET TEMP.

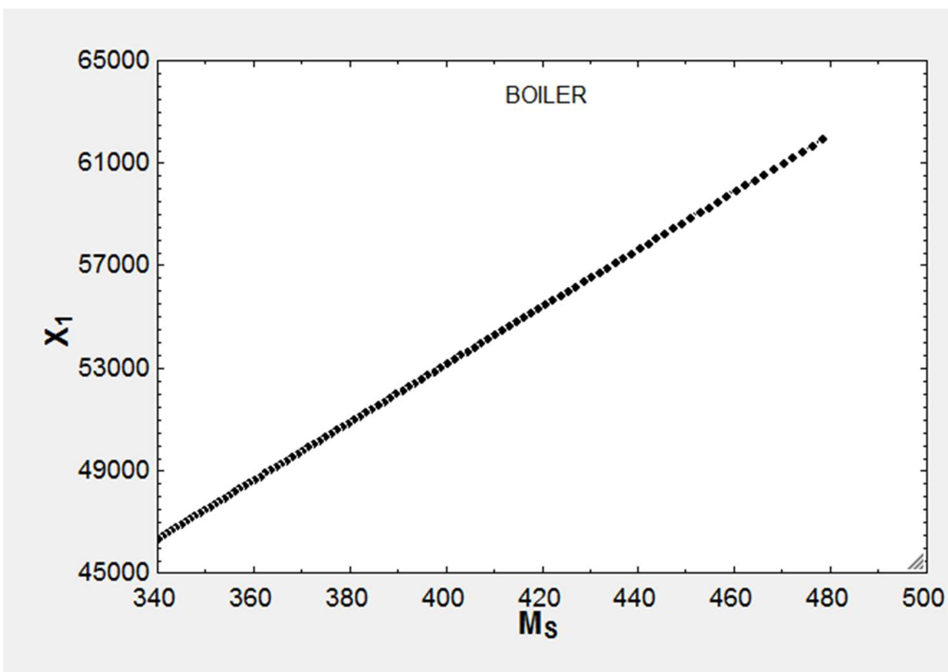


FIG 1.12 EXERGY LOSS IN BOILER VS. MASS FLOW RATE OF STEAM

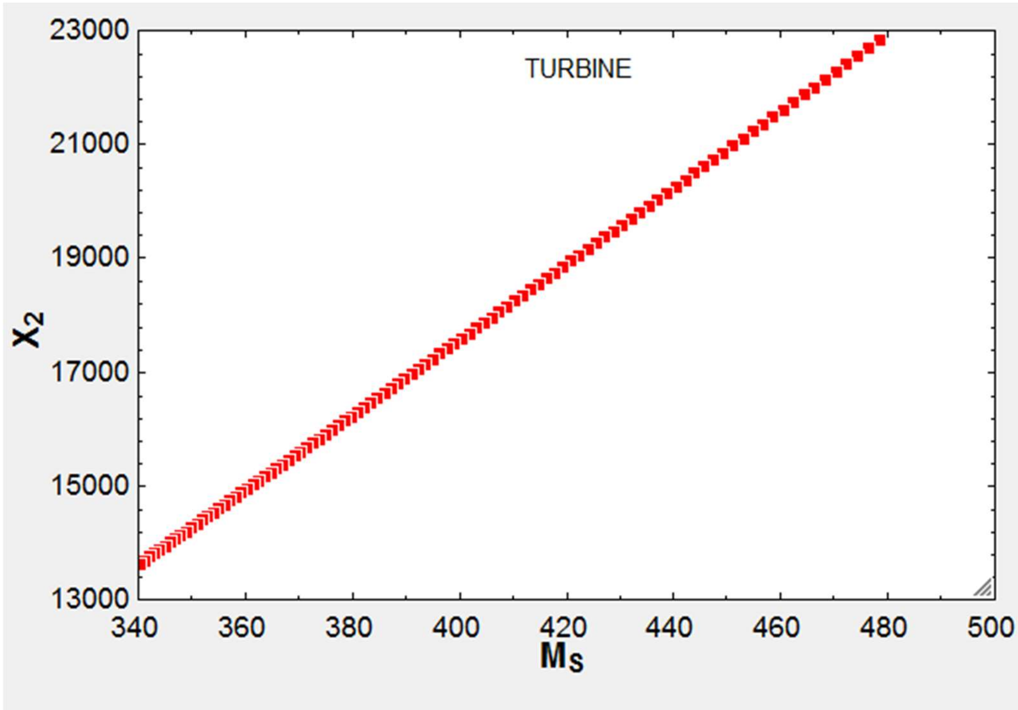


FIG 1.13 EXERGY LOSS IN TURBINE VS. MASS FLOW RATE OF STEAM

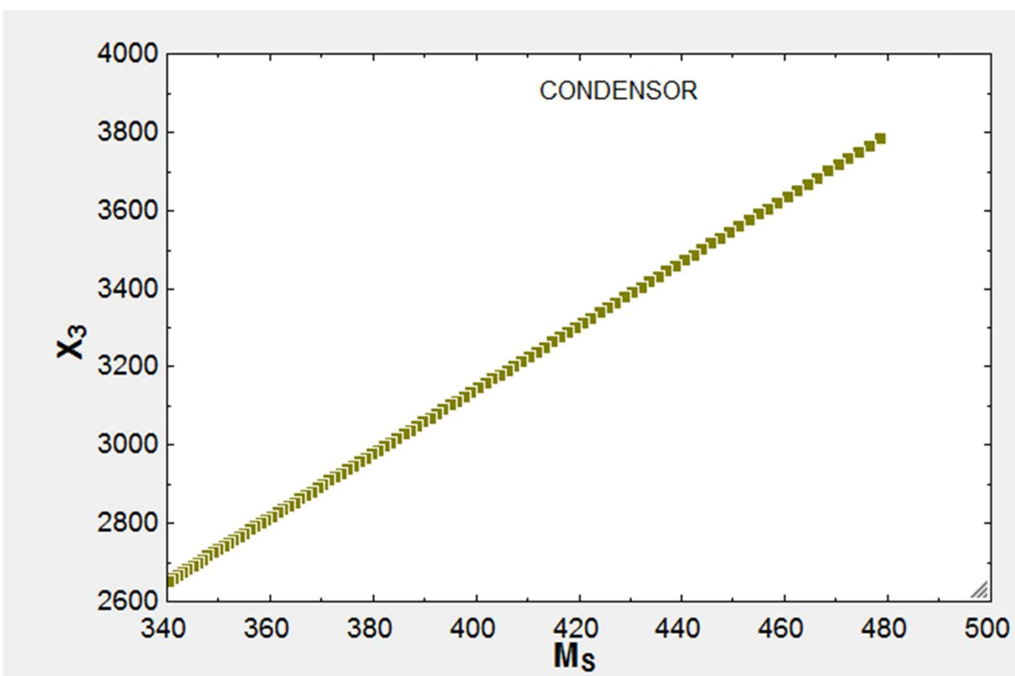


FIG 1.14 EXERGY LOSS IN CONDENSOR VS. MASS FLOW RATE OF STEAM

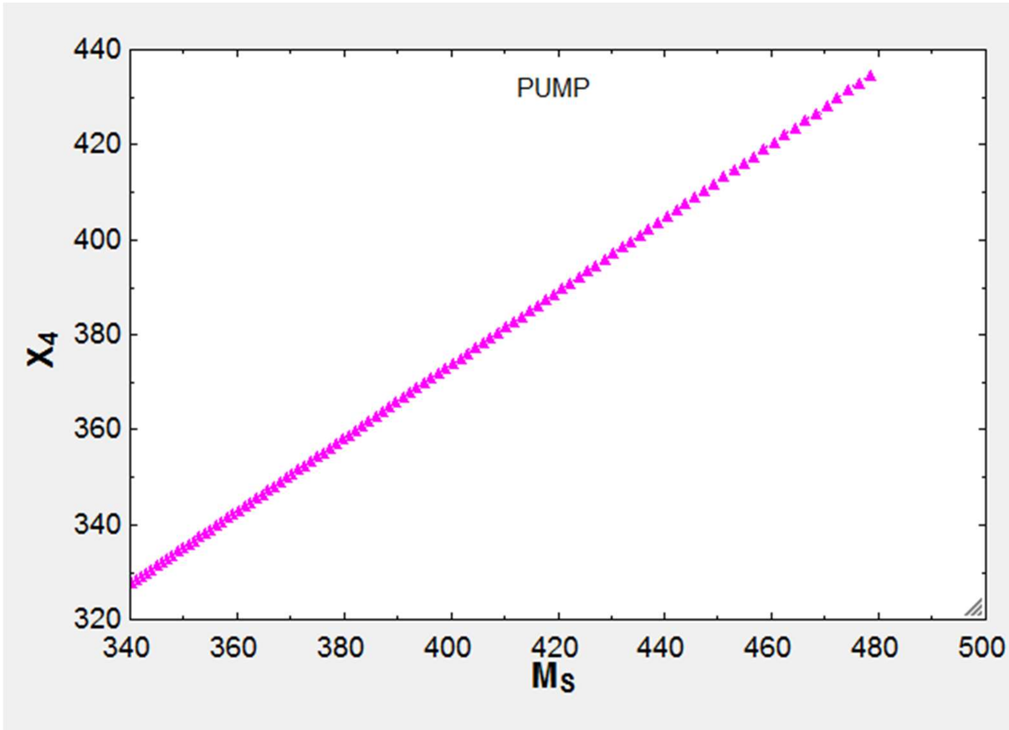


FIG 1.15 EXERGY LOSS IN PUMP VS. MASS FLOW RATE OF STEAM

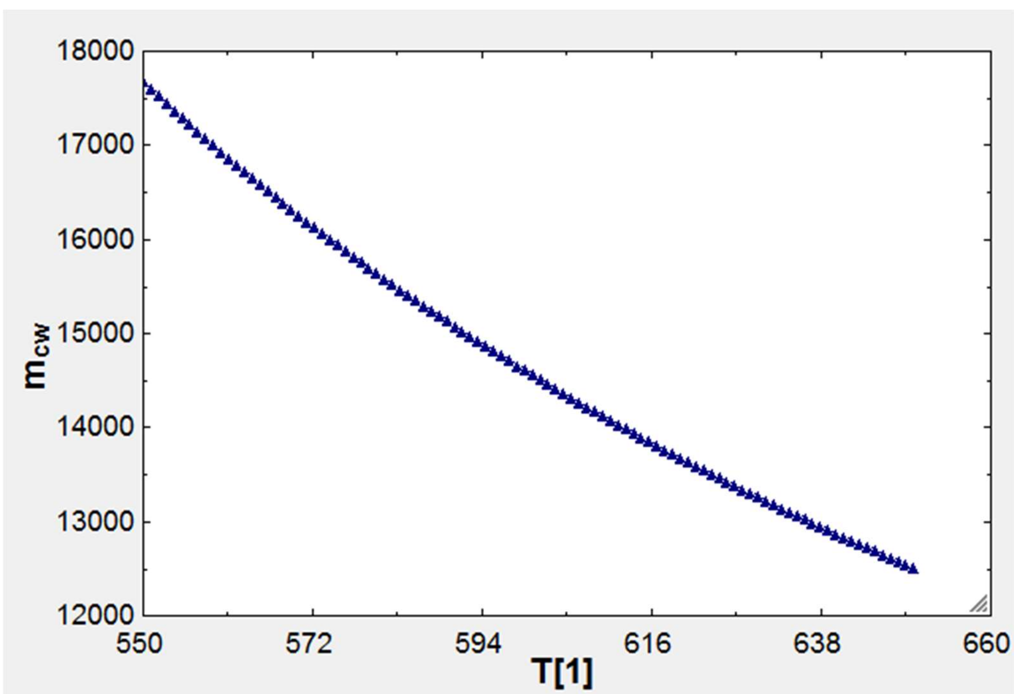


FIG 1.15 MASS FLOW RATE OF COOLING WATER VS. TURINE INLET TEMP.

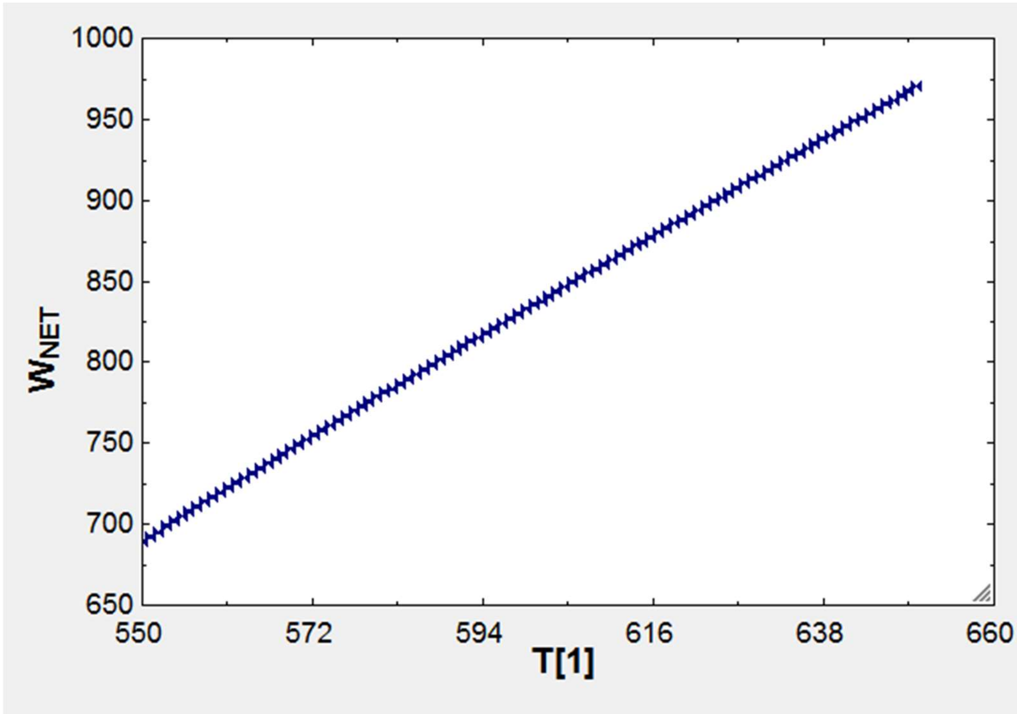


FIG 1.16 NET WORK OBTAINED IN CYCLE VS. TURBINE INLET TEMP.

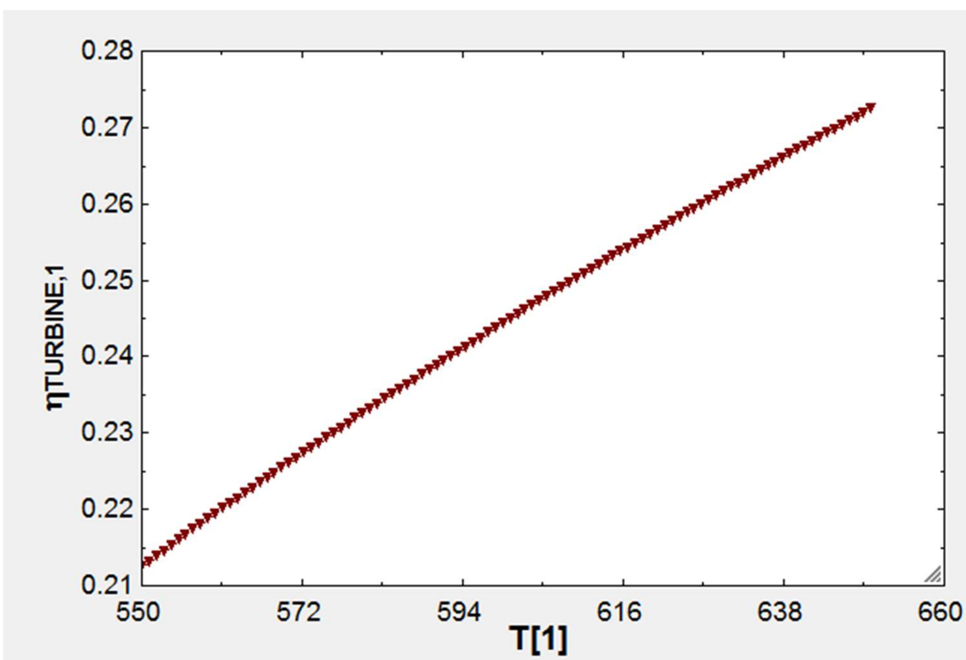


FIG 1.17 FIRST LAW EFF. OF TURBINE VS. TEMP. INLET TURBINE

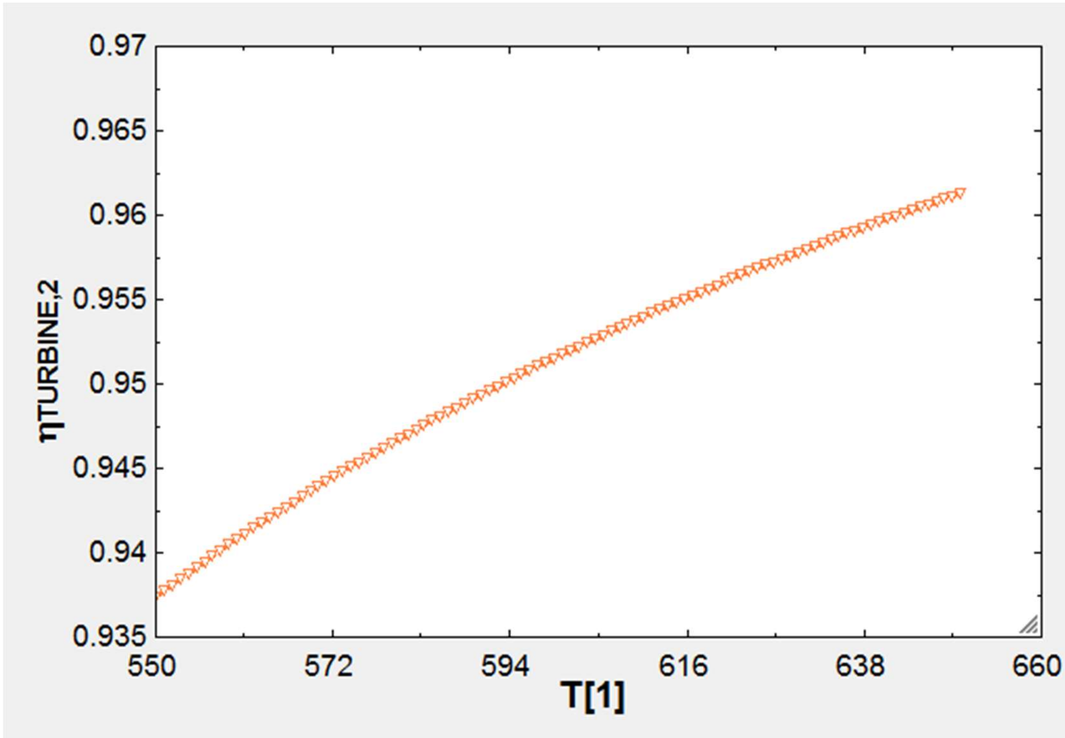


FIG 1.18 SECOND LAW EFF. OF TURBINE VS. TEMP. INLET TURBINE

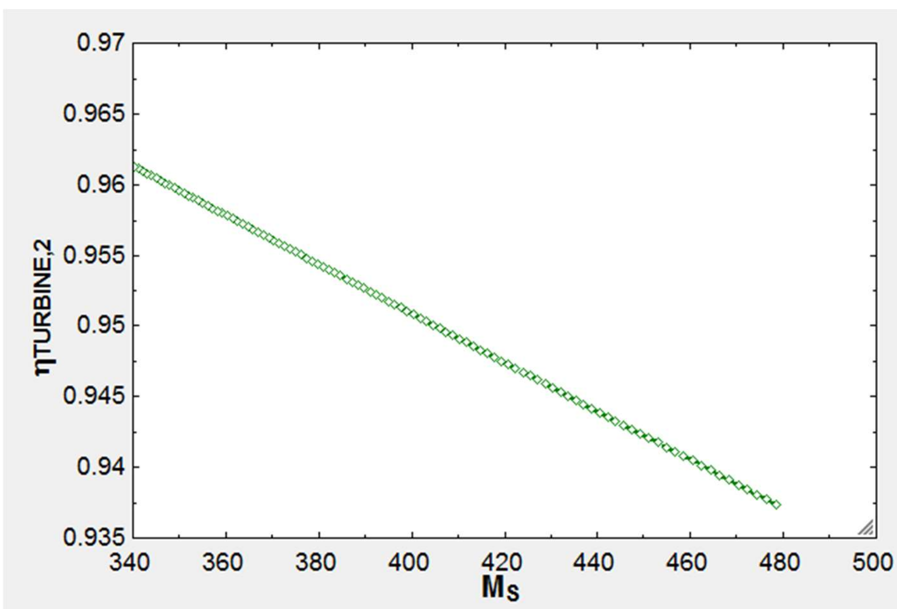


FIG 1.19 SECOND LAW EFF. OF TURBINE VS. MASS FLOW RATE OF STEAM

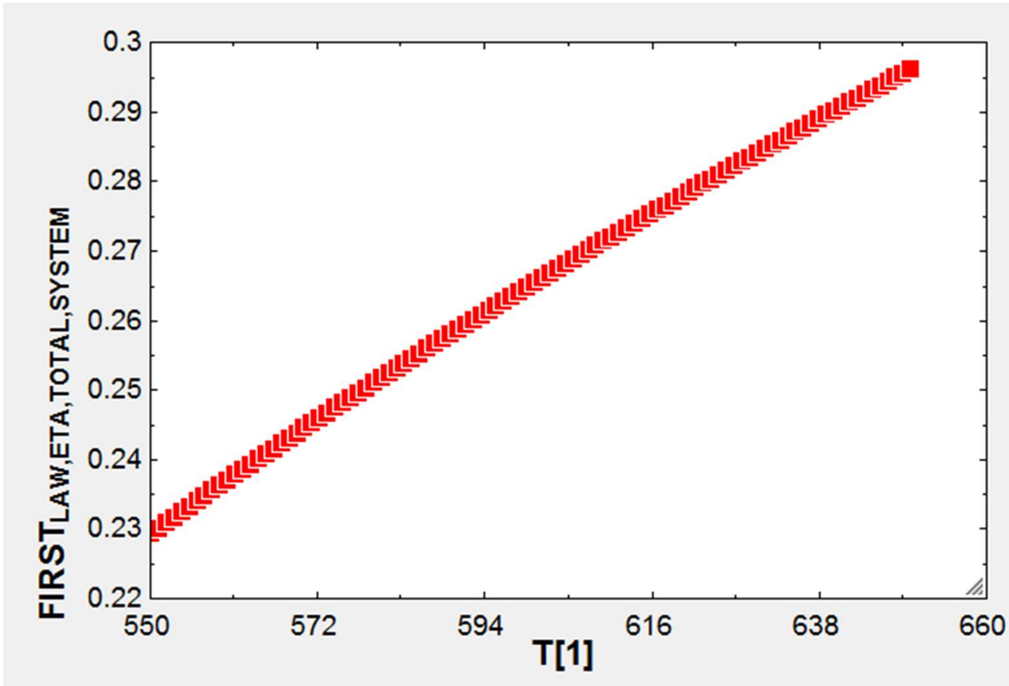


FIG 1.20 FIRST LAW EEF. OF CYCLE VS. TURBINE INLET TEMP.

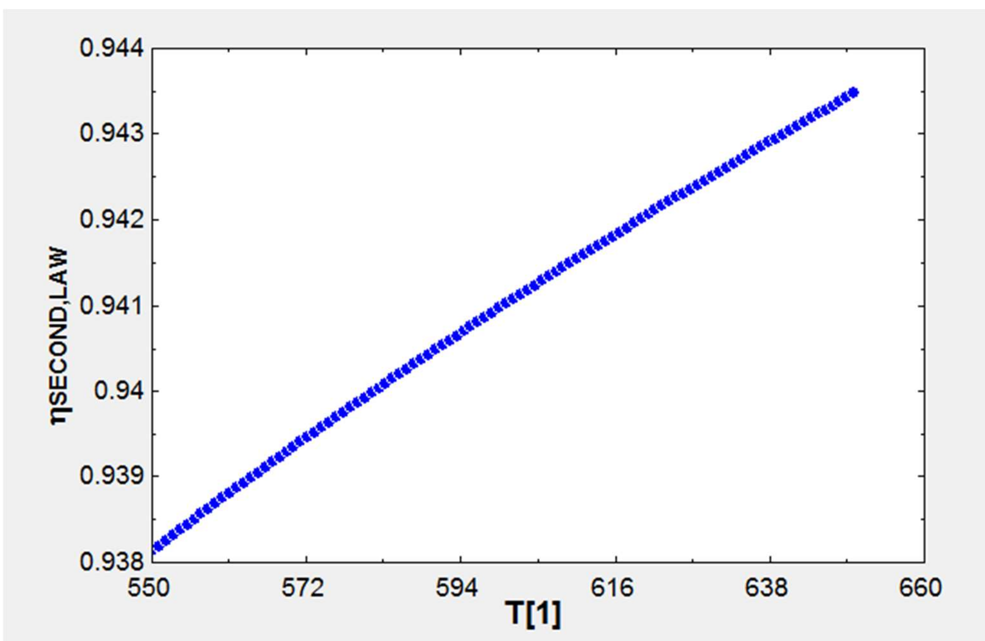
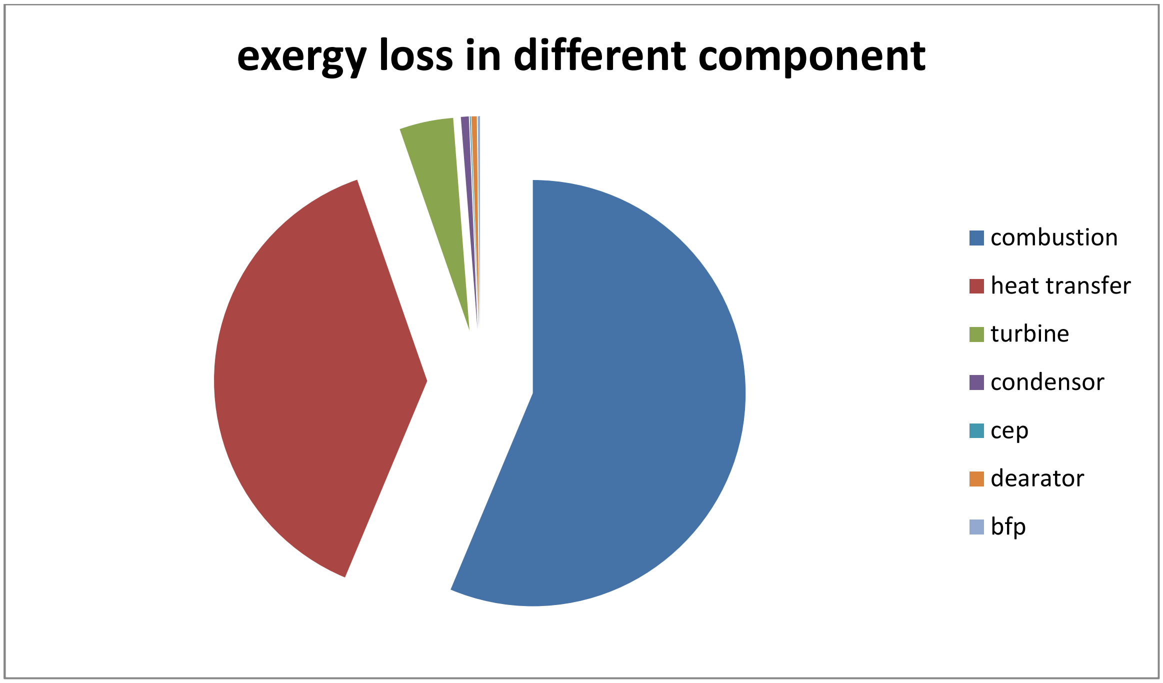


FIG 1.21 SECOND LAW EEF. OF CYCLE VS. TURBINE INLET TEMP.


exergy loss in different component
combustion
heat transfer
turbine
condensor
cep
dearator
bfp

.

FIG. 1.22

3.2 CONCLUSIONS AND FUTURE WORK

A second law of thermodynamic analysis was performed for a supercritical power plant to explore the performance of individual components. From the exergy analysis results obtained, the furnace was identified as having the highest exergy destruction. The turbines had the second highest exergy destruction rate.

Some of the conclusions regarding the exergy destruction of different components and options to reduce the exergy destruction in the furnace and turbine are given below:

1) Exergy destruction rate was greater in the combustor than the heat exchanger unit. The factors responsible for the furnace exergy losses were excess air percentage, preheated air temperature, and moisture content in the coal.

2) The exergy loss depends purely on the temperature range of the input and outlet stages of the turbine series. The exergetic efficiency increased in accordance with an increase in the reheating temperature.

The following are the recommendations for the future work:

For the power plant to operate at a very high temperature and pressure, it is important to focus more on the plant design. An advanced material is required to design a steam turbine that can withstand high temperature that is around 700 C.The steam turbine has to be designed in such a way that it matches the furnace conditions. Future research should also focus on identifying a coating that can withstand the high temperature and pressure and protect the equipment from steam oxidation and erosion inside the turbines. Also, the material properties throughout the overall design of an ultra supercritical power plant using coal as the fuel should be investigated.

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