Analysis of Thermal Power Plant and Design Modifications of its Components for improved Sustainability

A Thesis submitted in partial fulfilment of the requirements for the award of the degree of

DOCTOR OF PHILOSOPHY

by

Tarla Mallikharjuna Rao
(Old Roll No: 701007, New Roll No: 718084)

DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY
WARANGAL – 506004 TELANGANA
STATE, INDIA.

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under the supervision of

Prof. S. SRINIVASA RAO
Professor , Department of Mechanical Engineering
&
Dr K.T. Varughese
Former Additional Director,
Central Power Research Institute, Bangalore

DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY
WARANGAL – 506004
TELANGANA STATE, INDIA.
December– 2022

THESIS APPROVAL FOR Ph. D.

This thesis entitled **"Analysis of Thermal Power plant and Design Modifications of its Components for improved Sustainability"**by **Mr. TarlaMallikharjuna Rao**is approved for the degree of Doctor of Philosophy.

Examiners

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Supervisors

Dr. S. Srinivasa Rao
Professor
Department of Mechanical Engineering
NIT Warangal

Chairman Dr. V. Suresh Babu
Head, Mechanical Engineering Department,
NIT Warangal.

NATIONAL INSTITUTE OF TECHNOLOGY
WARANGAL – 506 004, Telangana State, INDIA

CERTIFICATE

This is to certify that the thesis entitled **“Analysis of Thermal Power plant and Design Modifications of its Components for improved Sustainability”** submitted by **Mr. TarlaMallikharjuna Rao**, **Roll No. 718084** , to **National Institute of Technology, Warangal** in partial fulfillment of the requirements for the award of the degree of **Doctor of Philosophy in Mechanical Engineering** is a record of bonafide research work carried out by him under our supervision and guidance. This work has not been submitted elsewhere for the award of any degree.

Dr. S. Srinivasa Rao
(Supervisor)
Professor
Department of Mechanical Engineering,
National Institute of Technology,
Warangal - 506004, Telangana State.

Dr. V. Suresh Babu
Chairman-DSC
Head of the Department
Department of Mechanical Engineering
National Institute of Technology
Warangal -506 004, Telangana, India.

NATIONAL INST ITUTE OF TECHNOLOGY
WARANGAL – 506 004, Telangana State, INDIA

DECLARATION

This is to certify that the work presented in the thesis entitled "**Analysis of Thermal Power plant and Design Modifications of its Components for improved Sustainability**", is a bonafide work done by me under the supervision of **Prof. S. Srinivasa Rao,** and was not submitted for the award of any degree to any other University or Institute.

I declare that this written submission represents my ideas in my own words and where ever others ideas or words are included have been adequately cited and referenced with the original sources. I also declare that I have adhered to all principles of academic honesty and integrity and have not misrepresented or fabricated or falsified any idea/data/fact/source in my submission. I understand that any violation of the above will cause for disciplinary action by the institute and can also evoke penal action from the sources which have thus not been properly cited or from whom proper permission has not been taken when needed.

Place: Warangal. **Tarla Mallikharjuna Rao**
Research scholar
Date: **Roll No: 718084**

Dedicated

to

Maa MAHALAXMI JAGDAMBA ,Koradi,Nagpur

My Family

My Teachers and friends who encouraged me

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For any errors or inadequacies that may remain in this work, of course, the responsibility is entirely my own.

(Tarla Mallikharjuna Rao)

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English

A cross sectional area,
d diameter,
f fin
g gravitational acceleration
h enthalpy
N total number of tubes
L length, m
m mass flow rate
P Pressure
Q Heat energy quantity
R resistance
r radius, m
c Specific heat
T Temperature
t thickness
U Heat transfer coefficient
u internal energy
p pressure
v volume
s tube pitch
S entropy
FD forced draft
ID induced draft
PA primary air
SA secondary air
APH Air pre heater
BFP Boiler feed pump
GCV gross calorific value
Bo bond number
LMTD log mean temperature difference
TPP Thermal power plant
NTU Number of transfer units (effectiveness of heat exchanger)
FW feed water
MS main steam
HRH Hot reheat header
CRH cold reheat header
HPT High pressure turbine
IPT Intermediate pressure turbine
LPT Low pressure turbine

Greek symbols

μ	dynamic viscosity, N,s. m^{-2}
ρ	density, kg.m^{-3}
σ	surface tension of liquid, N.m^{-1}
ε	Effectiveness
Ǝ	Exergy
η	efficiency
Δ	change in quantity
C	capital cost
Ə	endurability factor
ϒ	Exergo economic factor

Subscripts

a	adiabatic
b	thickness
c	condenser
D	destruction
e	evaporator
g	gas phase
l	liquid phase
h	height
TP	two-phase
p	constant pressure
L	coupling liquid
out	outside/outlet
in	inside/inlet
o	environmental
1	initial stage
2	final stage
r	rational
n	number
L	longitudinal
T	transverse
atm	atmospheric
sys	system
sat	saturated
FW	feed water
MS	main steam
HRH	Hot reheat header
CRH	cold reheat header
HPT	High pressure turbine
IPT	Intermediate pressure turbine
LPT	Low pressure turbine

Abstract

This study concerns to improve the performance of thermal power plants, so that the survivability of thermal power plants is improved amidst the present green power generation era. This is achieved by analyzing the thermal power plants more effectively through 5E analysis, i.e. energy, exergy, exergoeconomic, exergo-environmental and endurability analysis. This 5E analysis revealed that condenser, boiler, and turbine deserves improvement.

Various modifications such as, modifications of heat exchanger equipment by heat pipes and Solar blending of existing plants were discussed in this work.

The technology utilized to reduce exergy destruction is the use of successive heating and cooling in the heat exchangers employed in the power plant components such as super heaters, re heaters and condensers. This successive cooling and heating can be achieved by using heat pipes. Design details and calculations of the heat pipes which may be utilized in the super heaters, re heaters and condensers are presented in this work.

The thermal design modifications of platen super heater, final super-heater, re-heater and condenser were presented in this work and also it is proved by analytical analysis, reduction of exergy destruction in these components.

These thermal design modifications resulted the improvement of total plant efficiency from 31.09 % to 32.2%, exergy efficiency improvement from 49.7 to 53.5 %, exergo economic factor from 417.2 to 927.3, exergo environmental index from 18 % to 25.4 % and endurability factor from 0.689 to 0.581.

The schematic view of modified assemblies of platen super heater, final super heater and re heater are also shown. Similarly, modified condenser also presented. Also, the table top models of modified super heater and condenser prepared and presented in this work.

The heat pipes which are designed for condensation purpose are fabricated and experiments were conducted on the fabricated heat pipes. A laboratory model of the condenser with the heat pipes is fabricated and experiments were conducted on this heat pipe condenser.

*****

Chapter 1	Introduction

Chapter 1 Introduction

1.0 Introduction

Currently, the world is leaning towards green power generation such as Solar, wind power plants, bio fuel power plants, and hydro power plants. The reason for leaning towards this green energy generation is mainly due to the global warming and depletion of fossil fuels. Due to this, the existing thermal power plants are slowly being terminated and also installing a new thermal power plant is becoming difficult. But, it is a fact that in near future, the energy generation from thermal power plant (TPP) cannot be eliminated. The plants which are efficient, sustainable and least responsible for the global temperature rise will survive in the future.

Global warming influences the health, economics and comfort to the mankind and also effects the flora and fauna on the Earth. The various ill effects of global warming are uneven rains, severe cyclones, draughts and extreme climatic changes. But, the steady global warming is a natural phenomenon as there is difference in energy traffic of the Planet Earth in the Milky Way. But since the last century, there was a steep rise in the global warming especially after the industrial revolution. The main reasons are burning fossil fuels in inefficient manner, deforestation & tree cutting.

Earth is receiving energy from the Sun and it is being converted into various fuels. The fossil fuels are also energised from the Sun, but stored in the Earth crust for so many millions of years. Use of these fossil fuels triggers the exergy destruction and in turn entropy generation. It should be noted that entropy generated by burning of these fossil fuels is an addition to the already generating entropy by various natural process like photosynthesis, cloud formation etc. Now, the question is that where this extra generated entropy gets discarded or what is its effect.

The entropy generated is directly proportional to the exergy destruction. More entropy production indicates more exergy destruction, which in turn influences the global temperature rise. The temperature rise due to exergy destruction is 22.9 % of the global temperature rise in the year 2019 [1]. This temperature rise is only due to exergy destruction from the power plants as they are the major consumers of the fossil fuels.

Hence, the survivability of an existing TPP or an upcoming TPP merely depends on its sustainability, efficiency, cost effectiveness and gentle to the global atmosphere. For evaluating these parameters, it is vital to find the origin, causes and quantities of real losses in the system. Energy degrades during every conversion process [2]. Hence, it is essential to evaluate the energy quality along with the energy quantity for every TPP and also for all engineering process.

The performance of Thermal power plant, generally will be evaluated by First law of Thermodynamics and second law of thermodynamics. This, evaluation of TPPs are started with 1E (Energy) and progressed up to 4E (Energy, Exergy, Exergoeconomic, Exergeoenvironment) evaluation.

But the power plants are required to be evaluated more efficaciously i.e. more than 4E analysis so that the thermal power plants will become more sustainable and can continue in this green energy generation era.

1.1 Research Methodology

With above background, this project is carried out with the research methodology as shown in Fig 1.0

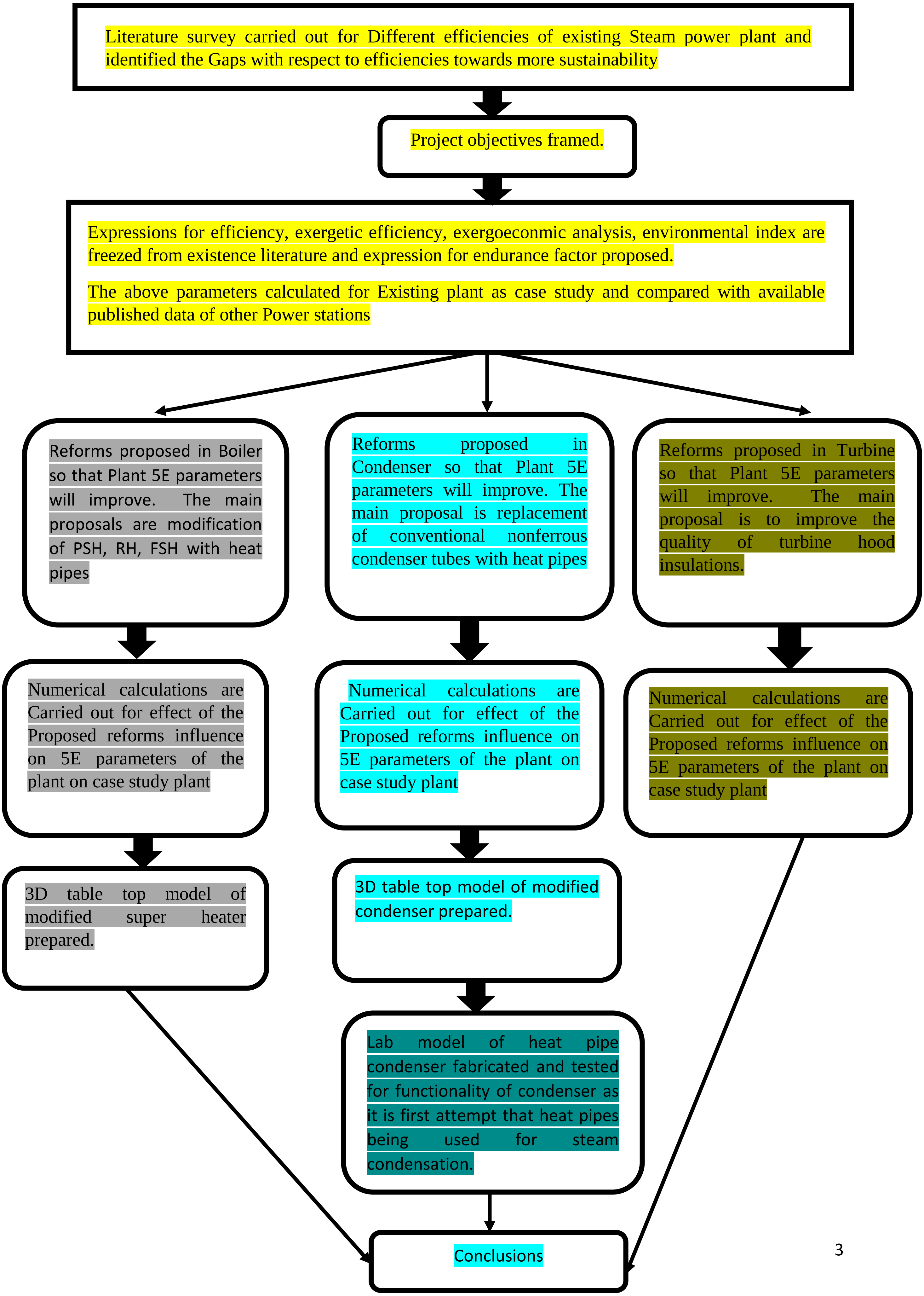


Fig 1.0. Research Methodology

Chapter 2

Literature Review

2.0. Introduction to Literature Review

The efficiency of the thermal power plants are evaluated as the ratio of the output energy to the input energy in the lines of first law of the thermodynamics and denoted as 1E evaluation. Here, the efficiency specifies the extent of efficient energy conversion. The energy loss of a power plant cannot be justified by the first law of thermodynamics, as it does not differentiate between the quality and quantity of energy. First law of thermodynamics is a simple energy balance without considering the quality of energy used. Exergetic analysis, based on second law of thermodynamics considers the energy quality. The energetic and exergetic analysis will provide a complete picture of the plant efficiency. Hence this exergy analysis getting popularity for the analysis of plants and engineering systems [3]. This exergy analysis is a more powerful tool than energy analysis for the power cycles because of the fact that it brings out the true magnitudes of losses, their causes and locations. Hence, in the recent decades the use of exergy analysis is found to be increasing and being accepted as a useful tool in the design, assessment, and optimization of energy systems. Thus 2E (Energy & Exergy) evaluation of power plants came into the picture and it gradually progressed up to 4E evaluation of power plants. The progress from 1E to 4E is surveyed and presented below:

2.1. 1E Analysis

Thermal Plants work on Rankine cycle, which was developed by William John Macquorn Rankine in 19th century [4]. It was suggested that the efficiency can be calculated as the ratio of work output to energy input. This is 1E analysis of a thermal power station. This efficiency calculation is based on first law of thermodynamics. Following this development, many researchers further worked on the efficiency improvement of thermal power plants by introducing reheating, regeneration etc.

2.2. 2E Analysis

1E analysis was as per First law of Thermodynamics and accounts the quantities of input and output energies. In this 1E analysis the sources of energy losses were not thought of [5, 6]. Then the concept of 2E analysis was introduced, i.e. the power plants can be analyzed by Energy and Exergy analysis [7]. The fundamentals of the Exergy method were laid by Saudi Carnot in 1824 and Rudolf Clausius in 1865.

Mehemet kanoglu, I Dincer et al [8] formulated the 2E analysis of power plants which helps engineers and policy makers for better use of energy & exergy efficiencies for energy handling of power plants. Sen Gupta et al. [9] analyzed a 210 MW coal plant exergetically and energetically and predicted that maximum exergy destruction was taking place in the boiler.

Isam H Aljundi [10] analysed power plant by 2E analysis and found that maximum exergy loss is occurring in the condenser. T Ganapathy et al. [11] published their findings which contains qualitative and quantitative analysis of a 50MW lignite burned power plant and concluded that maximum energy and exergy losses occur in the condenser and the combustor respectively. W.Zhang et al. [12] analyzed nine power plants with energy and exergy concept which helps to upgrade the performance of the plant components and also the entire plant. They identified the component with highest exergy loss and the influence of various parameters on the exergy efficiency.

Mitrovic D et al. [13] analyzed thermal plants, energy wise and exergy wise and concluded that energy losses are more in condenser and exergy destruction is large in boiler. P Regulagadda et al.[14] analyzed a sub critical operated plant by energy and exergy formulations at different operating parameters and concluded that boiler and turbine irreversibility cause maximum plant irreversibility.

Y Y Jiang et al. [15] studied power plants by 2E method and suggested that the increase of feed water temperature and secondary air temperature increases the exergy efficiency of the boiler. Kaushik et al. [16], Ahmadi and Toghraie[17] analyzed coal fired thermal power plants and concluded that the energy losses occur mainly in condenser (almost 66 % of energy input) while the highest exergy losses occur in the boiler. Mali Sanjay et al [18] analyzed 125 MW coal based thermal power plant by 2E analysis and concluded that 47.43 % exergy loss occur in the combustor and it needs modification.

2.3. 3E Analysis

A significant number of investigators worked on 2E (Energy & Exergy) analysis of thermal power plants and concluded that major exergy losses occur in boiler and condenser. Then, another E (exergoeconomic analysis) crept into the evaluation of power plant analysis.

Mohammad Ameri et al. [19] worked about 3E (energy, exergy and exergoeconomic) analysis of a 250 MW thermal power plant and concluded that the maximum energy loss occurs in the condenser and next to it in the boiler. It was also concluded that the cost of exergy destruction in the boiler and turbine is higher than any other component of the power plant.

Ali Bolatturk et al [20] carried out 3E analysis of Cayirhan thermal power plant and concluded that in a Thermal power plant, highest amount of exergy losses are witnessed in boiler, turbine, condenser and heater groups. The cost of exergy loss in boiler, turbine group and condenser are in decreasing order of their magnitude.

Ulna F, ozkan D.B [21] analyzed thermal power plants by 3E analysis and utilized the exergoecomic factors to improve the plant efficiency. They concluded that the energy loss, the exergy loss and the cost loss of exergy destruction of boiler, turbine group, condenser, heater group, pumps group and auxiliary group are in decreasing order of their magnitude. Mohammad H. Ahmadi et al. [22] published a review on the comprehensive analysis of energy, exergy economic (3E) analysis and their application related to various thermal power plants. It was concluded that in coal fired power plants, highest level of energy losses occur in the condenser and boiler.

2.4. 4E Analysis

With respect to increase in global warming and the influence of the pollutants from the thermal power plants on this global warming, the concept 4E (energy, exergy, exergoeconomic and exergoenvironment) analysis evolved for the thermal power plants evaluation. Marc A Rosen and I Dincer [23] published their work and concluded that exergy analysis has an important role in increasing green energy.

Ameri, M et al. [24] worked on the 4E analysis of a large steam power plant as a case study in the year 2016. The following conclusions have been drawn.

i. The exergy destruction in the boiler and turbine are about 86 and 8% of total exergy destruction of power plant respectively.
ii. The excess air has an impact on exergy efficiency
iii. With improvement of efficiency and exergy efficiency , correspondingly power generation and environmental costs decrease.

Ravinder Kumar [25] made a critical review on energy, exergy ,exergoeconomic and economic (4-E) analysis of thermal power plants and concluded that major exergy destruction occurring in the boiler and cost of exergy destruction in the boiler and turbine is higher in comparison to other components cost.

G Dattarao , Sarath Babu et al. [26] worked on 4E analyses of chemical looping in subcritical and super critical plants . It was concluded that chemical looping combustion based sub critical and super critical plants are energetically, exergetically, ecologically and economically efficient.

Kabiri et al. [27] analysed an old steam power plant with 4E analysis that is energy , exergy, exergoeconmic and exergoenvironmental analysis and concluded that , after modifying the plant to combined cycle, the plant resulted increase in efficiency, decreased greenhouse emissions and a 57 % drop in exergy destruction costs.

Muhammad Faizan Tahir et al [28] made a comprehensive review of 4E analysis of coal and gas fired thermal power plants. It was concluded that exergy analysis is a reasonable index to complement energy analysis by computing losses to environment and internal losses or exergy destruction. It was also concluded that, Exergoeconmic and Exergoenvironmental analysis indicates that system which has least exergy destruction, economically competitive and ecologically benign.

2.5. 5E Analysis

According to the above literature, the thermal power plants had been evaluated by 1E, 2E, 3E and 4E analysis by number of investigators. But these 4E analysis are not enough for the survival of thermal plants in the forthcoming green energy production era. Hence it is proposed that the thermal power plants are to be evaluated by 5E analysis that is Energy, Exergy, Exergoeconomic, Exergoenvironmental and Endurability analysis for better evaluation of thermal power plants and no open literature is available for this 5E analysis.

2.6. Observation from the literature

- The evaluation of Thermal power plants started from 1E analysis and progressed up to 4E analysis.
- Boiler, Turbine and condenser are major sources of exergy destruction and hence low exergetic efficiency.
- As energy efficiency and exergy efficiency increases, correspondingly the generation and environmental costs decreases. Exergy analysis is a reasonable index to complement energy analysis by computing losses to environment and internal losses or exergy destruction.

2.7. Gap in the Evaluation of Power Plant Efficiency

It is noticed from the literature review that the following areas are not much concentrated:

- ➢ Literature review reveals the progress of evaluation of power plants. But these 4E analysis are not enough for the survival of thermal plants in the forthcoming green energy production era. Hence to evaluate the thermal power plants in more compatible way, there is gap in the evaluation the plant in terms of endurance (Endurance is the ability of plant to produce the power with minimum exergy destruction so that the plant will be less harmful to environment).

- ➢ Limited work has been carried out for improving the 4E efficiency parameters of the TPP.

- ➢ Not much work done for design modification for improvement of exergy efficiency and improvement of 4E parameters of power plant.

2.8. Conclusion and Objectives for the present work

Thermal power plants which burn fossil fuels like coal, for electric power generation are evaluated by 4E analysis only. But to survive these plants in near future, these plants are to be evaluated by 5E analysis, so that the plants will become economically competitive and ecologically benign. Enough information is not available the literature for this 5E analysis. Hence the present work carried out with the following objectives.

(i) 5E analysis formulation for evaluating efficiency, exergetic efficiency, exergoeconmic analysis, environmental index and endurance factor for a typical power plant. Numerical calculations of these 5E parameters for a typical operating coal based thermal power plant as a case study.

(ii) Proposal of technologies to improve the 5E parameters of the Thermal power plants, so that plants become environmentally viable and its theoretical evaluation using an operating TPP as case study.

(iii) Thermal and physical design of heat pipes to suit application in super heater, re heater and condenser which are vital components of a TPP.

(iv) A laboratory model is to be fabricated with one of above proposed concept and experiments are to be conducted on the laboratory model to prove that the suggested method can be successfully applicable to TPP application.

***

Chapter 3	5E Analysis formulation and calculations for a Typical Power Plant

3.0 Description of Steam Power plant

The modern steam power plants, whose major components are boiler, turbine, generator and condenser, are based on the Modified Rankine cycle which is presented in Fig. 3.1. In this cycle, the water is pumped inside boiler through a BFP and it is converted into super-heated steam through different stages of boiler with the help of coal combustion. This super-heated steam drives turbine, which in turn rotates the generator to produce the electrical energy [29].

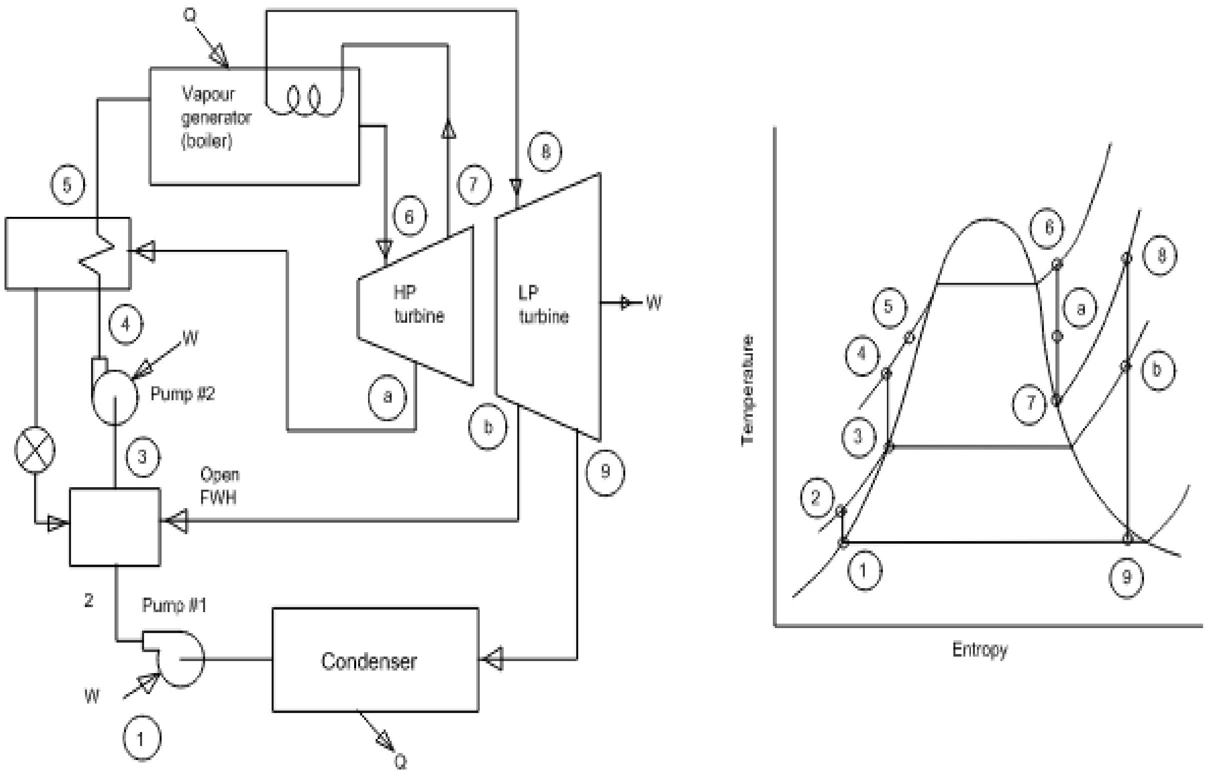


Fig 3.1. The typical steam power plant with TS diagram

3.1 5E analysis formulation

First and second law of thermodynamics are basis for evaluation of a TPP. This, evaluation of TPPs was started with 1E (Energy) and progressed up to 4E (Energy, Exergy, Exergoeconomic, Exergeoenvironment) .

3.1.1 Energy analysis: [30]

$$\eta = \text{Work Output} / \text{Energy input}.$$

3.1.2 Exergy analysis: [31,32]

The exergy of a system ,Ǝ, at a specified state is given by the expression

$$Ǝ = (U-U_0)+ p_0(V-V_0) - T_0(S-S_0) + KE+ PE$$

Exergy balance for a closed system can be evaluated by,

$$\Delta Ǝ = change\ in\ exergy = \int_1^2 \left(1-\frac{T_0}{T_b}\right)\delta Q - [W - p_0\,(V_2 - V_1)] - T_0\Delta S$$

3.1.3 Exergoeconomic Analysis:[33]

$$\Upsilon = Ǝ_{out}\,(Ǝ_{\eta\ 1}/1-Ǝ_{\eta\ 1})$$

3.1.4 Exergoenvironmental Analysis: [34]

$$\text{Exergy Renewability index, } Ǝ_{RI} = \frac{\Sigma\, Ǝ_{Products}}{Ǝ_{Fuel} + Ǝ_{destroyed} + Ǝ_{deactivation} + Ǝ_{disposal} + \Sigma Ǝ_{emission}}$$

3.1.5 Endurability Analysis (Proposed in this Work)

Almost all the thermal systems are supplied the exergy inputs derived directly or indirectly from consumption of fossil fuels. Every engineering process inevitably consists the exergy loss. This exergy consumption, which is known as exergy destruction will play a big role in the sustainability of fuels. More is the exergy destruction, less is the energy sustainability and vice versa. This endurability factor will be calculated as,

$$\text{Endurability factor} = Ə = \text{Exergy Destruction} / \text{Exergy input}$$

3.2 Calculation of 5E for a case Study

Based on the formulae 3.1 to 3.5, the 5E are calculated for a case study power plant. A 210 MW TPP, which is functioning in India was considered as case study. (Even though 500 ,660 ,800 MW plants are now available, as an initial step 210 MW plant considered at the moment , which can be extended to higher MW plants) .This TPP working on Rankine cycle with reheating.

The Boiler of this plant operating with pulverized coal firing burners, super heater, re heater, economizer etc.).Turbine consists three sub stages viz. high, intermediate and low pressure turbines. Shell and tube type surface condenser used for condensing turbine exhaust steam. The detail parameters of power plant, considered are given in Tables 1, 2, 3 in the Appendix I. The related calculations are presented in Appendix II & Appendix III.

The 5E parameters for the above described power plant were calculated, according to the formulae described and presented in the Table 3.1. These parameters compared with available literature for the similar plants.

TABLE 3.1. 5E PARAMETERS OF EXISTING PLANT

E description	Value calculated for the case study	Reference Number				
		11 (T Ganapathy et al.)	7 (S C Kaushik et al.)	14 (P Regulagadda et al.)	35 (A Rashad et al.)	36 (S Sengupta et al.)
1E (efficiency)	31.09 %	27 %	NA	30.12 %	44.8%	37%
2E (exergy efficiency)	49.7 %	27%	26.95%	25.38 %	43.9%	36%
3E (Exegoeconomic factor)	417.2	NA	NA	NA	NA	NA
4E (Exergoenvironmental index)	18 %	NA	NA	NA	NA	NA
5E (Endurability Factor)	0.689	NA	NA	NA	NA	NA

(NA – Not available)

3.3 Conclusion for this chapter

The formulae of 1E, 2E, 3E and 4E are collected from the literature and used in this work. The formula for 5E is suggested in this work and is utilized for calculating endurance factor of the thermal power plants.

The Table 3.1, indicates that, the not only 5E parameter not available the power plants and also 3E, 4E parameters are not calculated for most of power plants.

Close observance of the Table 3.1 implies that there is a lot of scope to improve these 5E parameters.

*****

Chapter 4	Proposals to increase 5E parameters

4.0. Introduction to Proposals

The survivability of an existing thermal power plants largely depends on these 5E parameters. Also for installation of the new plant amidst competition from green energy plants, the new plant should have better 5E parameters and low carbon footprint. Though significant contributions have been made in studying the exergy destruction, exergo-environmental effects etc., the design modifications proposed to improve these characteristics are muted creating a void in the literature. This thesis proposes few technology modifications (listed in Table 1) to improve the 5E parameters.

Table 4.1. DIFFERENT PROPOSALS TO REDUCE EXERGY DESTRUCTION SO THAT 5E PARAMETERS IMPROVE

Sl. No	Component	Technique to reduce Exergy Reduction
I	Condenser	Successive cooling of steam is proposed to decrease the exergy destruction. Heat pipes are proposed to achieve this of cooling i.e. replacement of conventional non- ferrous tubes by specifically designed heat pipes.
II	Boiler	1. Successive heating of steam is proposed in platen, final and re heater coil assemblies to reduce exergy destruction. This will possible by replacing tubes with heat pipes. 2. Introduction of steam blending with Solar energy. 3. Improving the boiler skin insulation properties
III	Turbine	Exergy destruction reduction is proposed by improving the turbine insulation properties.

The above proposals with their feasibility are discussed below. The discussions are conducted by considering a case study TPP.

4.1 Proposals to decrease exergy destruction in Condenser

4.1.1 Description of the Existing Condenser

Condenser is an vital equipment in TPP. Most of power plants use surface type condenser consisting shell an tubes. The heat transfer mechanism of this type condenser is shown in Fig 4.1.

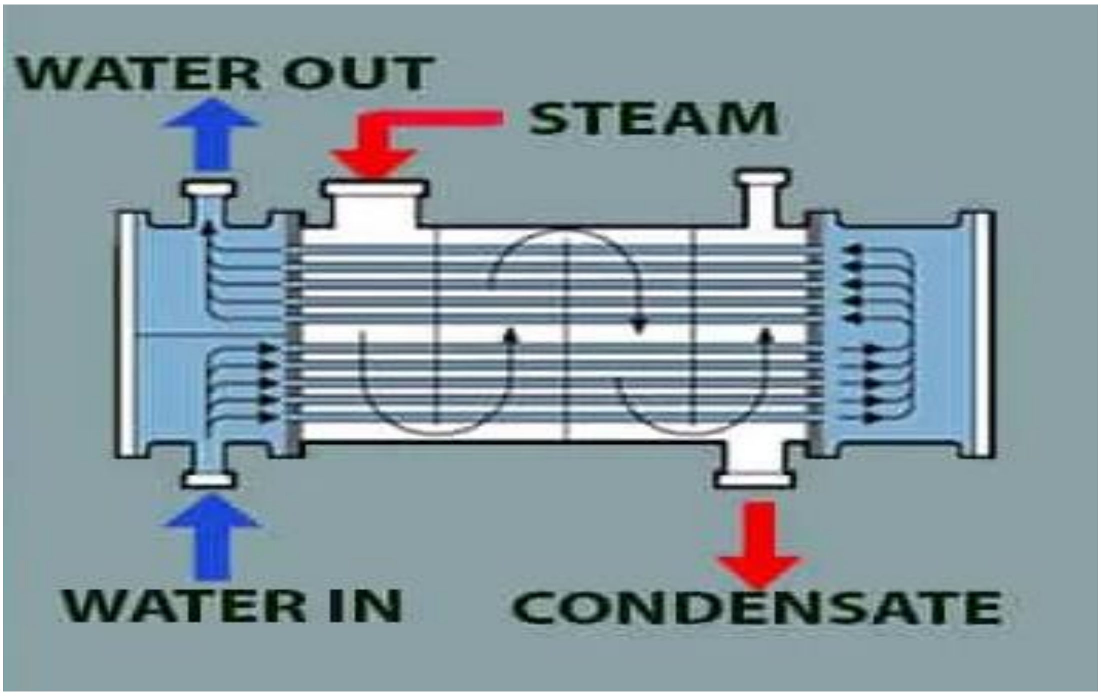


Fig 4.1 Heat Transfer mechanism in existing condenser

The steam from the turbine exhaust surrounds the tube surface of the condenser. Because of cold water flow inside condenser tubes, the tube surface temperature is maintained below the saturation temperate of surrounded steam. This results in the steam condensation. The heat transfer took place between steam and cooling water inside condenser tubes. Steam released heat energy by virtue of latent heat of condensation and cooling water absorbed heat energy by virtue of sensible temperature rise. Due to this reason, large area of heat transfer is required which necessitates more number of tubes. As a result, the number of tubes drastically increases as the capacity of power plant increases. As a consequence the operation and maintenance problems increases.

The exergetic efficiency of the existing condenser is calculated as below.

(The detailed calculations are presented in Appendix II)

4.1.2 Exergy Calculation of the Existing Condenser

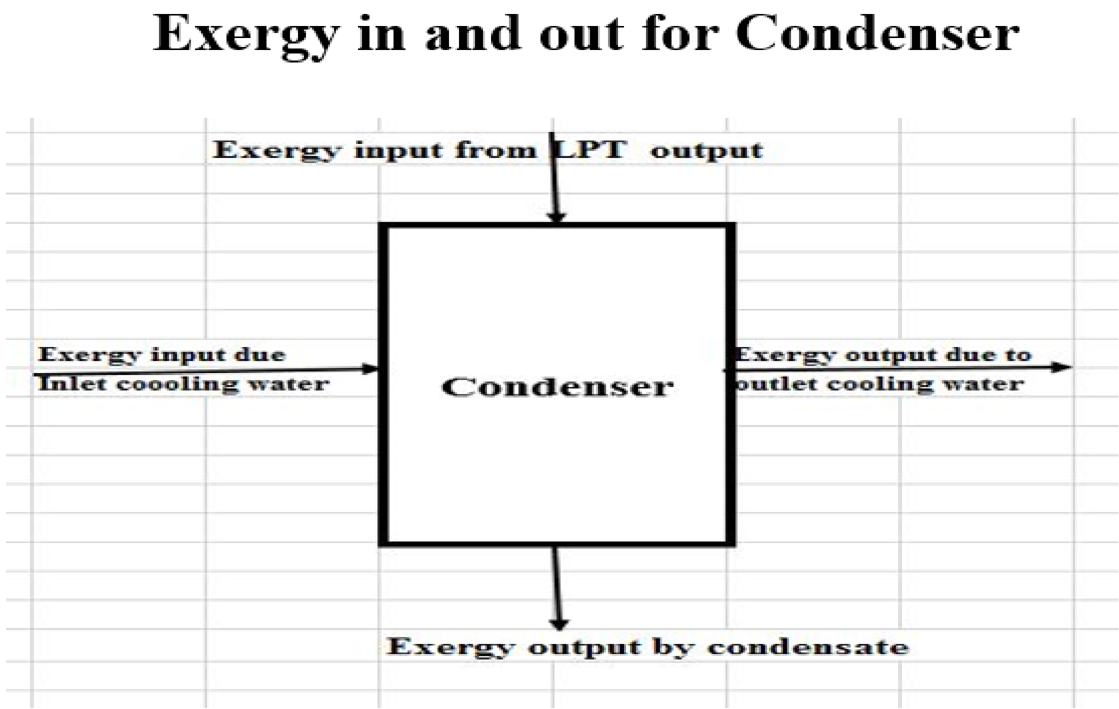


Fig 4.2 Exergy traffic for the condenser

The formulae used for exergy calculations are,

Exergy inlet due to exhaust steam from LP Turbine = $\exists_{cond,\ s,\ in}$

$$= \dot{m}_{steam,in} [(h_{steam,in} - h_o) - T_o(S_{steam,in} - S_o)]$$

Exergy entry due to cold water is $\exists_{cond,\ w,\ in}$

$$= \dot{m}_{cold,,in} [(h_{water,,in} - h_o) - T_o(S_{water,in} - S_o)]$$

Exergy destruction in the condenser = $\exists_{cond,d}$

$$= T_{env} [\ CW \ln (T_{cl,out} / T_{cl,in}) + CW (T_{cl,in} - T_{cl,out})/T_{s,in}\ ]$$ [38]

Exergetic Efficiency of the Condenser = $\frac{Exergy\ inlet - exergy\ outlet}{Exergy\ inlet}$

Applying the numerical values in the above equations,

- Exergy input to the condenser due to steam =14.3 MW.
- Exergy entered by the cooling water into the condenser = 0.62 MW.
- Exergy destruction in the condenser = 11.65 MW.
- Exergetic Efficiency = Second law of Thermodynamic efficiency=22%.

4.1.3 *Probable reasons for exergy destruction*

- Direct Heat Transfer between steam and cooling water, where the temperature difference between two streams are high.
- Exhaust condensate losses.

4.1.4 Proposal to improve the exergetic efficiency of condenser and reduce the number of cooling tubes

As stated in Table 4.1, the reduction in exergy destruction is possible by successive cooling of the turbine exhausted steam. That is, the steam is not directly cooled by cooling water. The heat energy from the steam is transferred to an intermediate liquid and then that absorbed heat energy from the intermediate liquid is transferred to the cooling water. The method is presented in Figure 4.3.

Fig No 4.3. Successive cooling of Steam

To achieve the phase change of cooling water and successive cooling of steam, the heat pipes may be considered for steam condensation. For this purpose, a suitable heat pipe is designed to suit the requirement as shown in Fig. 4.4. The design details are given in Table 4.2. The dimensions of heat pipe have been arrived based on area availability at the existing plant. (The design calculations are presented in Appendix IV) .The thermodynamics limits of the proposed heat pipe are listed in Table 4.3.

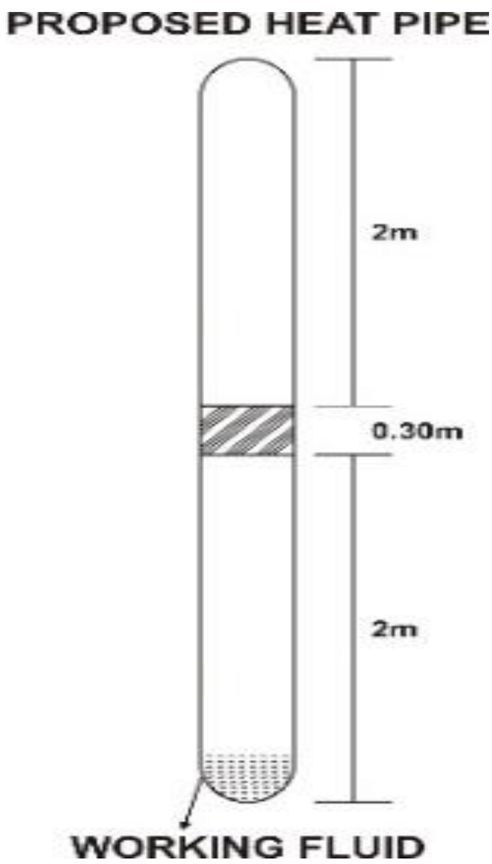


Fig 4.4. Proposed Heat Pipe

TABLE 4.2. SPECIFICATIONS OF HEAT PIPE

Parameter	**Numerical value**
Length	4.3 m
Material	Copper
Vacuum inside heat pipe	0.07 bar
Working fluid	Distilled water
Working fluid saturation temperature under vaccum	39.02 ° C
Wick material	Wickless heat pipe

TABLE 4.3. THERMODYNAMIC PROPERTIES OF DESIGNED HEAT PIPE

Parameters	Desired requirements of Heat Pipes in the proposed HPHE	Designed Heat pipes characteristics
Maximum heat transfer limit from the Boiling point of view	30 kW	71 kW
Maximum heat transfer limit from the Flooding point of view	30 kW	59.3 kW
Overall heat transfer coefficient under prevailing conditions		2447

With the above specified heat pipe , the condenser proposed. The details of the proposed condenser with the above heat pipes are presented in Table 4.4. The line diagram of laboratory model of is shown in Fig 4.5. The related calculations are presented in Appendix V.

TABLE 4.4. PROPOSED HEAT PIPE BASED CONDENSER

Parameter	Value
Total number of heat pipes	9025
Arrangement of Heat pipes in HPHPE	Staggered , 95 x 95
Load on each Heat Pipe and heat transfer rate per unit area	28.8 kW ≈ 30 kW and 84.73 kW/m^2

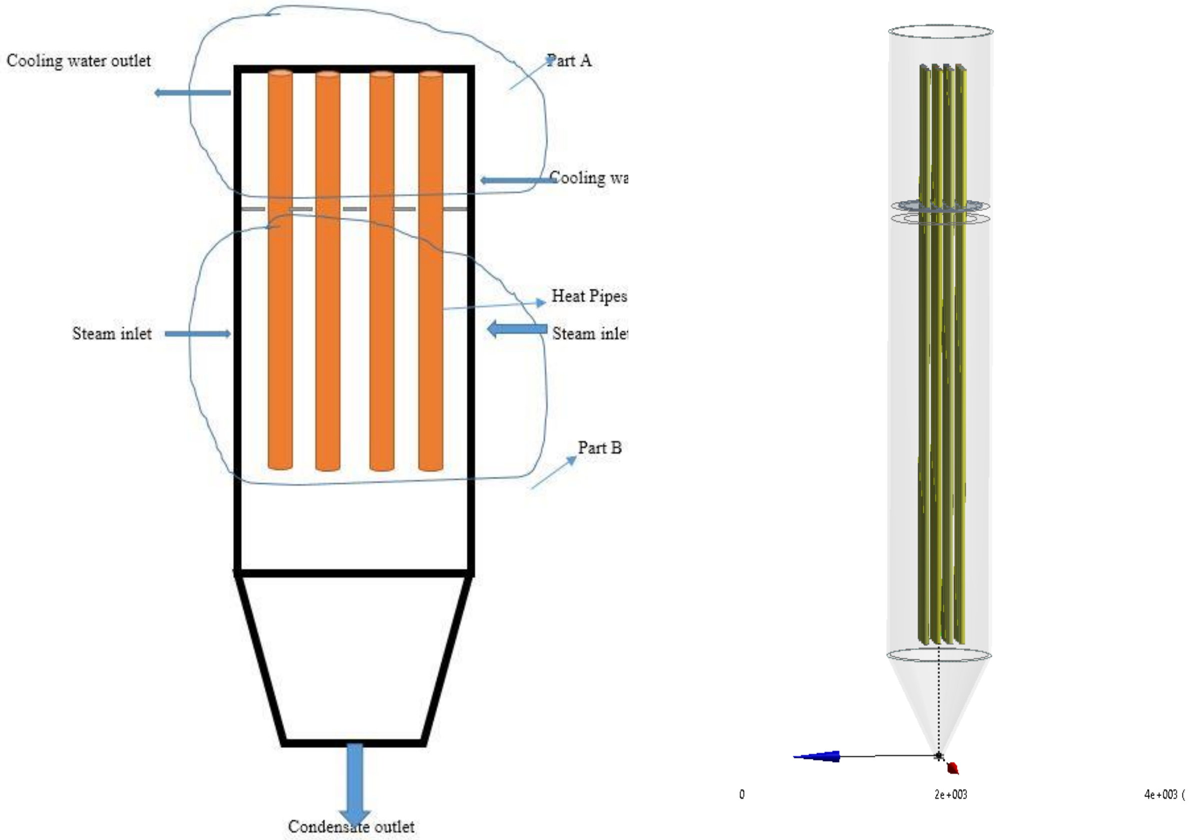


Fig 4.5: Line diagram of Proposed Condenser

Steam inlet conditions into the condenser = 46 ° C, 0.09 bar (The values are from the running power plant, which is considered as case study)

Condenser load for 210 MW turbine, = 221171743.8 kW ≈ 260 MW

Proposed Heat pipes in the condenser = 9025

Based on the calculations in Appendix 5,

Heat Transfer Coefficient (h_1) for the portion of heat pipe which is exposed to inlet Steam = $1.8 \times (9.43 \times 10^{16})^{0.25}$ = 31,543 w/m^2.k

Heat Transfer Coefficient (h4) for the portion of heat pipe which is exposed to water Pool = $h_4 = k \times Nu / d_o$ = 21,405 w/m^{2}K

Heat Transfer Coefficient (h_2) for the evaporator section, inside of heat pipe = 15,521 w/m^{2}k

Heat Transfer Coefficient (h_3) for the condenser section, inside of heat pipe = 14,065 w/m^2.k

Overall heat transfer coefficient for single heat pipe = U = 2406 $w/m^2.k$

Based on the formula, $Q = U_p NA\ (LMTD)$, the number of heat pipes required are 6158. This indicated that 6158 number of heat pipes serves the purpose. However, to take care of dirt factor and unexpected heat load fluctuations it is proposed to have 9025 tubes.

4.1.5. Comparison between conventional condenser and HPHE condenser.

i. Effectiveness Comparison

Effectiveness of the Existing Condenser = Actual Heat Transfer/Max. Possible Heat Transfer [42]

Effectiveness of conventional condenser = 55 %

Calculation of the proposed heat pipe heat condenser Effectiveness is adopted from ref. [42,43] as liquid-coupled, indirect-transfer-type exchanger and calculated based on expression, $\varepsilon_o = \{ 1/\sum\varepsilon_e + (C_{hl}/C_{cl}) \times (1/\sum\varepsilon_c - 1) \}^{-1}$,since $C_{cl} > Cx_l > C_{hl}$

Appling numerical values, Effectiveness of Proposed Heat pipe based condenser ≈ 0.99

The above values indicates that proposed condenser is better than existing condenser.

ii. Heat Transfer per unit area

The heat transfer area comparison between two condensers are listed in Table 4.5 with a condensing load of 260 MW.

TABLE 4.5. COMPARISON OF TWO TYPES OF CONDENSER

Parameter	**Conventional Condenser**	**Heat Pipe Based Condenser**
Number of Tubes	19208	9025 Heat pipes
Diameter of tubes	0.0254 m	0.0540 m
Length of tubes exposed for steam condensation	11.28 m	2 m
Total Heat Transfer area *(Length x Perimeter x Number of Tubes)*	17289.2 m^2	3062.1 m^2
Heat Transfer Rate	15 kW/m^2	85 kW/m^2

iii. Exergy analysis

Exergy destruction of Conventional Condenser as calculated in 4.1.2, Ǝ = 11.65 MW

and Exergetic efficiency = 22 %

<u>Exergy destruction of proposed Heat pipe condenser</u> (Detailed calculations presented in Appendix V)

. As shown in Fig 4.5, steam enter in Part A and cooling water enters in Part B and condensate flows from the bottom. As per reference [38] the calculations of exergy are carried out and presented below.

For Part B, where steam condenses into water and fluid inside the heat pipe evaporates.

$$\Delta Ǝ = U_o A_o(\pi_T - 1)^2/\pi_T$$

For Part A, where cooling water gets heated and vapor inside heat pipe condenses into liquid.

$$\Delta Ǝ = T_{env}[C_1 \ln(T_1''/T_1') + C_1(T_1' - T_1'')/T_2']$$

Total Exergy destruction in Part A and Part B = 5.92 MW

And Exergetic efficiency = 61%

The above calculations indicates that use of heat pipes in the condenser reduces the exergy destruction and hence exergetic efficiency improves by 39%.

The exergetic efficiency of heat pipe condenser is compared with exergetic efficiencies of other power plant condensers available in the published data and presented in Fig 4.6.

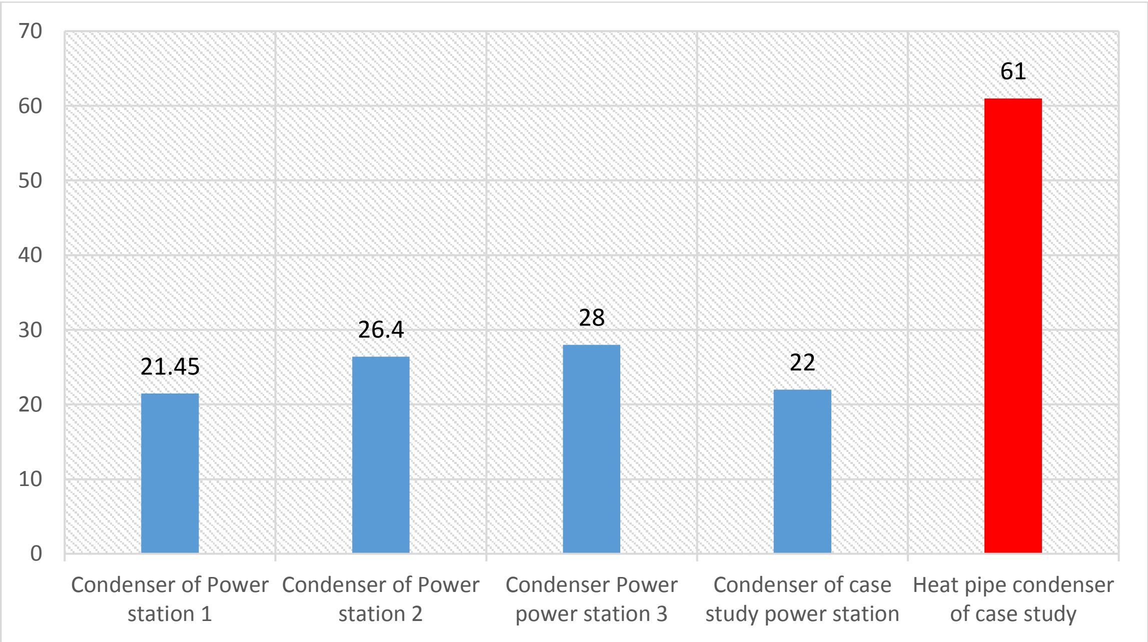


Fig 4.6. Comparison of exergetic efficiencies of condensers with HP condenser

4.2 Proposals to increase 5E parameters in Boiler

Sengupta and A Datta [9] reported that maximum energy destruction in a power plant occurs in the boiler (about 60%) and the following reasons could be attributed for the decrease in exergy efficiency.

4.2.1 Exergy Calculation of the Existing boiler

The Exergy input and output in a boiler will be as shown in Fig 4.7

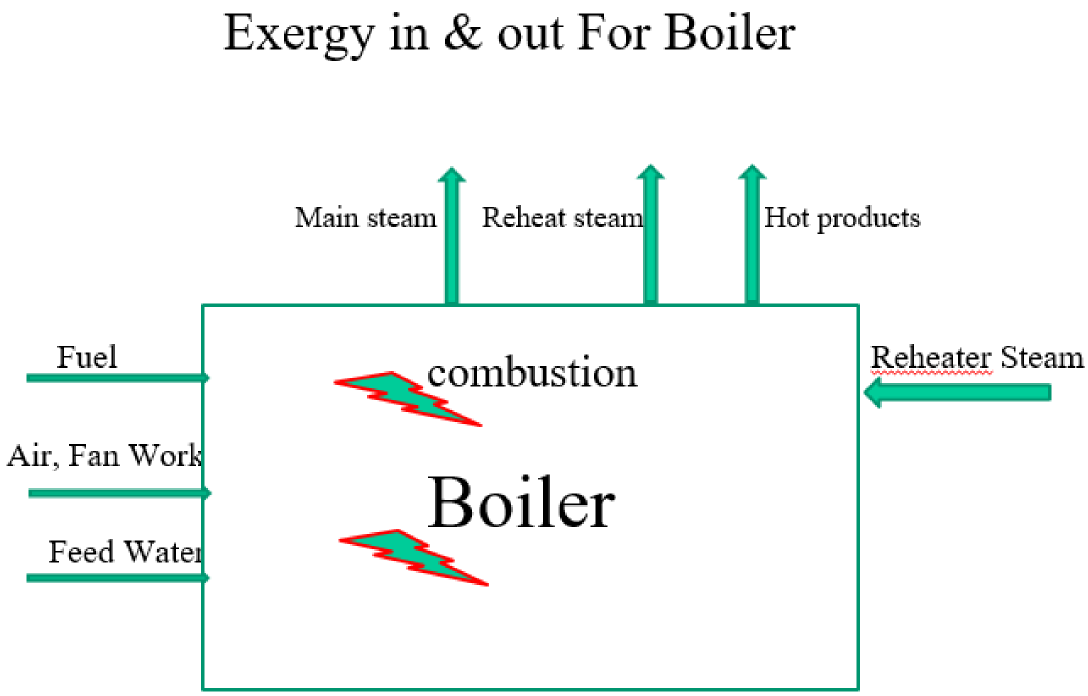


Fig 4.7 Exergy traffic for the Boiler

. Exergy Calculation Existing Boiler (The calculations are presented Appendix II)

A. <u>Exegy inlet into the Boiler $\exists_{B,in}$</u>

i. Exergy supplied by the Fuel coal , $\exists_{Fuel}$

= Coal GCV x Coal Firing Rate x Exergy grade factor of fuel

= 621.3 MW

ii. Exergy supplied by Feed water, $\exists_{FW}$

$= \dot{m}_{FW}\,(CP)_{FW}.\,[(T_{FW}-T_0)-T_0\ln(T_{FW}/T_0)]$

= 43.7 MW

iii. Exergy Due to inlet Air, $\exists_{A}$

Air from atmosphere enters APH through FD fan. APH is considered as part of boilers. Hence it is assumed the inlet air doesn't carry any exergy into the boiler. The outlet of APH is divided into Primary Air (PA) and Secondary Air (SA).

The exergy inlet is due to FD and PA fan work

The work input due to 2 FD fans and 2 PA fans = 4 MW

iv. Exergy due to Re-heater inlet (CRH) , $\exists_{CRH}$

$$= \dot{m}_{CRH}[(h_{CRH}-h_0) - T_0(S_{CRH} - S_0)]$$

$$= 174.2 \text{ MW}$$

Total Exergy input to boiler, $\exists_{B,in}$

$$\exists_{B,in} = \exists_{Fuel} + \exists_{FW} + \exists_{A} + \exists_{CRH} + \text{BFP work}$$

$$= 850.2 \text{ MW}$$

B. **<u>Exegy Exit of the Boiler $\exists_{BE}$</u>**

i. Exergy carried out by Main steam, $\exists_{MS}$

$$\exists_{MS} = \dot{m}_{MS}[(h_{MS} - h_o) - T_o(S_{MS} - S_o)]$$

$$= 251.7 \text{ MW}$$

ii. Exergy carried out by HRH, $\exists_{HRH}$

$$\exists_{HRH} = \dot{m}_{HRH}[(h_{HRH} - h_o) - T_o(S_{HRH} - S_o)]$$

$$= 191.6 \text{MW}$$

iii. Exergy carried out by Hot Products , $\exists_{HPr}$

$$\exists_{HPr} = \dot{m}_{HP}[(h_{HP} - h_o) - T_o(S_{HP} - S_o)]$$

$$= 6.8 \text{ MW}$$

Now , Total Exery exit of the boiler,

$$\boldsymbol{\exists_{B,out}} = \exists_{MS} + \exists_{HRH} + \exists_{HP}$$

$$= 251.7 + 191.6 + 6.8$$

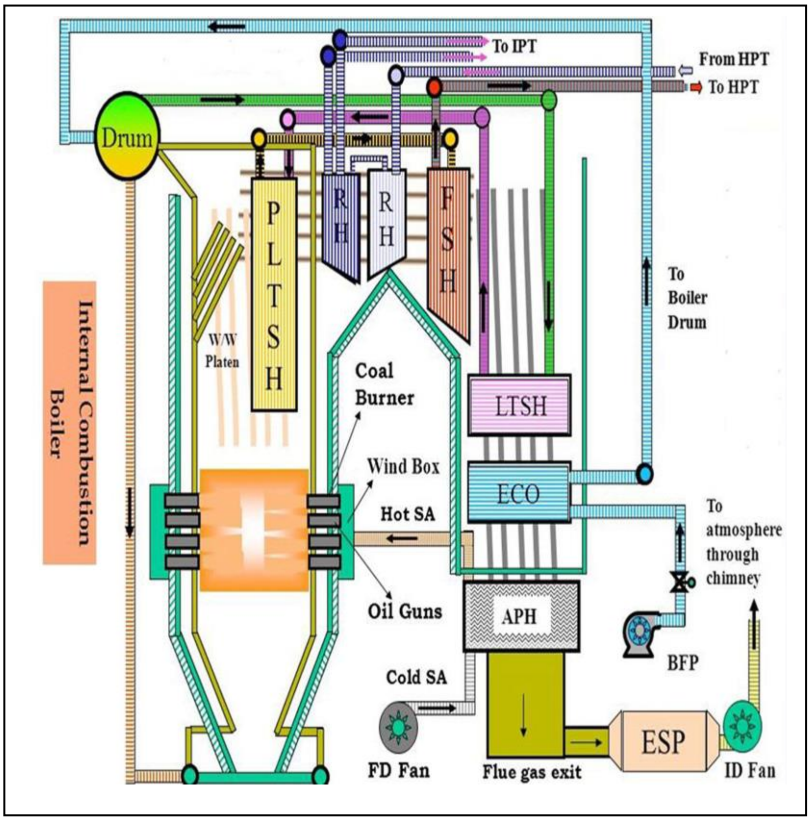
$$= 450.1 \text{ MW}$$

Exergy destruction in the boiler = $\boldsymbol{\exists_{B,D}}$ = 850.2-450.1 = 400.1 MW

Exgergy efficiency of boiler = $\boldsymbol{\exists_{\eta B}}$ **=** 450.1/850.2 = 52.9 %

4.2.2 *Most Probable Reasons for Large Exergy Destruction*

The reasons of exergy destruction in the boiler is due to heat Transfer between the high temperature gases and the relatively low temperature steam (as shown in Fig 4.8), Exhust flue gas losses, heat energy losses from boiler skin to environment and Combustion losses



Component Name	Inlet/ Outlet	Gas Temperatures in Deg C	Steam Temperatures in Deg C
Water Wall Tubes	Intlet	1300	286
	outlet	1500	404
Platen Super Heater	Inlet	1236	404
	Outlet	1077	475
Final Super Heater	Inlet	1077	475
	Outlet	962	535
Reheater	Inlet	962	330
	Outlet	724	535
LTSH	Inlet	724	359
	Outlet	481	404
Economiser	Inlet	481	243
	Outlet	328	286

Fig 4.8. Heat transfer Pattern in the present type of boiler (210 MW Plant boiler)

4.2.3 *Proposals to reduce exergy reduction in the boiler*

i. The boiler outer walls are covered by insulation pads. These pads are designed According to ASTMC 680 standards [37] to maintain a surface temperature of 60°C. (the detailed calculations are in Appendix VI)

Total boiler outside wall surface area = 1800 m^2 (for case study boiler)

Considering, average boiler inside temperature = 1000 ^{o}C

The heat lost from the boiler outside surface walls $= \frac{\Delta T}{\Sigma R} = \frac{1000-60}{0.325/1800}$ = 5.2 MW

Hence the exergy transferred to environment = [1- 298/(273+60)]x 5.2

= 0.546 MW

It is proposed, if insulation properties are increased, so that the if, boiler wall surface is maintained at 40°C, then exergy transferred will be as follows.

The heat lost from the boiler outside surface walls $= \frac{\Delta T}{\Sigma R} = \frac{1000-40}{0.325/1800} = 5.32$ MW

Hence the exergy transferred to environment = [1- 298/(273+40)]x 5.32
= 0.255 MW

Improvement in Exergy destruction saving $= \frac{0546-0.255}{850.2} = 0.03$ %

ii. The reduction of exergy destruction in boiler is proposed by improving performance of plant heat transfer equipment like platen super heater, final super heater and re heater. The successive heating successive heating (Fig 4.9) of steam is proposed to reduce exergy destruction and hence the exergy efficiency improvement.

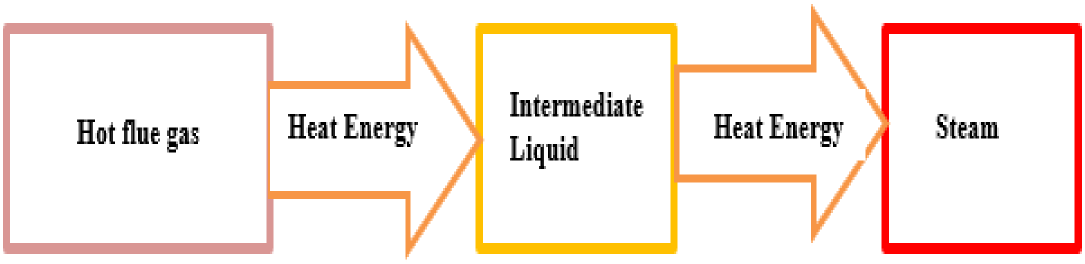
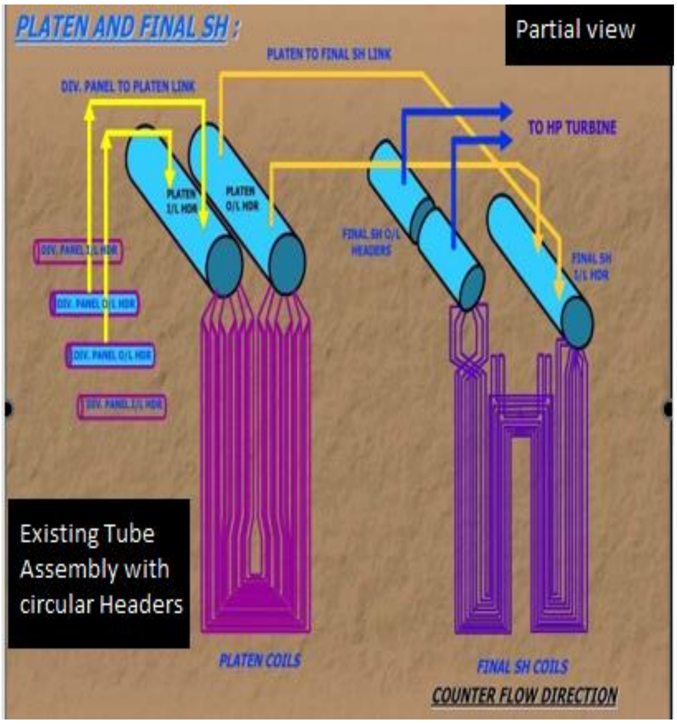
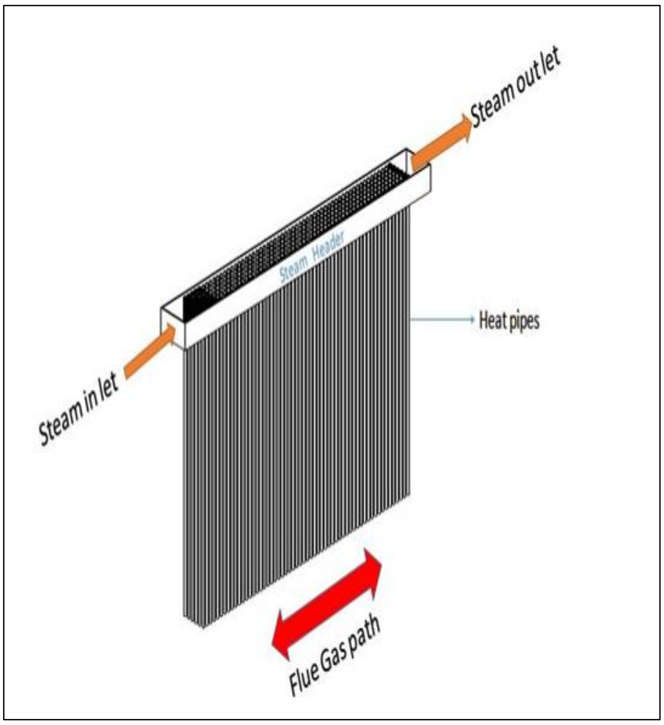


Fig 4.9 Successive Heating of Steam

Based on this concept of successive heating, the design of platen super heater may be modified as shown in Fig 4.10. In this proposed modifications the bunch of heat transfer coils will be replaced with suitable heat pipes. As a consequence, the circular header will be replaced with a rectangular header.





Existing Assembly Proposed Assembly

Fig. 4.10. Proposed Modification in Heat transfer system of boiler

The line diagram and specification of the proposed heat pipes for intended job are given Fig 4.11. The dimensions of heat pipe have been arrived based on area availability at the existing plant. The thermodynamic details are given Table 4.6.

Heat pipe Details

- Length = L = 13.05 m
- Length of evaporator section = L_e = 12 m
- Length of the condenser section = L_c = 1 m
  (This section of heat pipe consists fins)
- Length of the adiabatic section = l_a = 0.05 m
- Material is 316L SS tubes
- Vacuum inside heat pipe = 0.15 bar
- Working fluid is Sodium metal. The proper ties are as follows.
- The Fin Details
- N_f = Number of fins = 200 fins/m = n
- Thickness of fin = b = 3mm = 0.003 m
- The fin height = h = 25 mm = 0.025 m
- Outer Diameter of tube = d=d_o = 0.090 m
- Base diameter of fin = outer diameter of tube = d=0.090 m
- S_L = S_T = Tube pitch = 190 mm = 0.19 m
- S = Clearance between fins = (1/n - b) = 1/200 – 0.003 = 2 x 10^{-3} m
- Flue gas inlet temperature = 1236 ° C as per boiler collected data.
- The overall heat transfer coefficient of heat pipes, U = 166.7 W/m^2 K

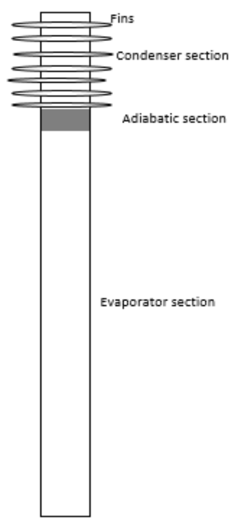


Fig 4.11 Line diagram and specification of Proposed Heat Pipe

TABLE 4.6. THERMODYNAMIC DETAILS OF PROPOSED HEAT PIPES.

Sl.No	Parameters	Desired requirements of Heat Pipes in the proposed HPHE	Designed Heat pipes characteristics as per different calculations
1	Maximum heat transfer limit from the Boiling point of view	130 kW	766 kW from (3)
2	Maximum heat transfer limit from the Flooding point of view	130 kW	155 kW from (4)

The Calcualtion for heat pipes and their suitability to Platen Super heater are presented in Appendix VII.

Heat Transfer Coefficient for the portion of heat pipe which is exposed to inlet

Flue gas = 262.5 $w/m^2.K$

Heat Transfer Coefficient for the evaporator section, inside of heat pipe
= 853.13 kw/m^2K

Heat Transfer Coefficient for the condenser section, inside of heat pipe
= 3724.7 $kw/m^2.K$

Heat Transfer Coefficient for the portion of heat pipe which is exposed to Steam

= 1255.83 w/m^2K

Overall heat transfer coefficient 166.7 w/m^2K

Based on the formula**,** $Q = U_pNA$ (LMTD) , the number of heat pipes required are 120 tubes. This indicated that 120 number of heat pipes serves the purpose. However, to take care of dirt factor and unexpected heat load fluctuations it is proposed to have 348 tubes.

The finned tubes are proposed to arrange as 6 x 58 (to accommodate in the existing space), that is 6 tubes in width and 58 rows in depth in the direction of steam flow. The height of fin tube is 1 m. The fins are of SS 316 L material same as tube material.

Pressure Drop across the finned tube bundle that is in the steam is calculated on the Formula = $[0.205(f+a)G^2N_d]/\rho_{steam}$ and numerical value obtained is 0.405 bar. Whereas, in the conventional header the pressure drop is 3 bar.

As a result of two phase heat transfer mechanism of heat pipes the heat transfer coefficient improves significantly and hence the number of tubes required for heat duty will decrease . In addition, because of successive heating of steam the exergy destruction will reduce. The proposed assembly to platen super heaer is shown in Fig

4.12. Similarly the proposed assemblies for final super heater and reheater are shown in Fig 4.13 and Fig 4.14 respectively .

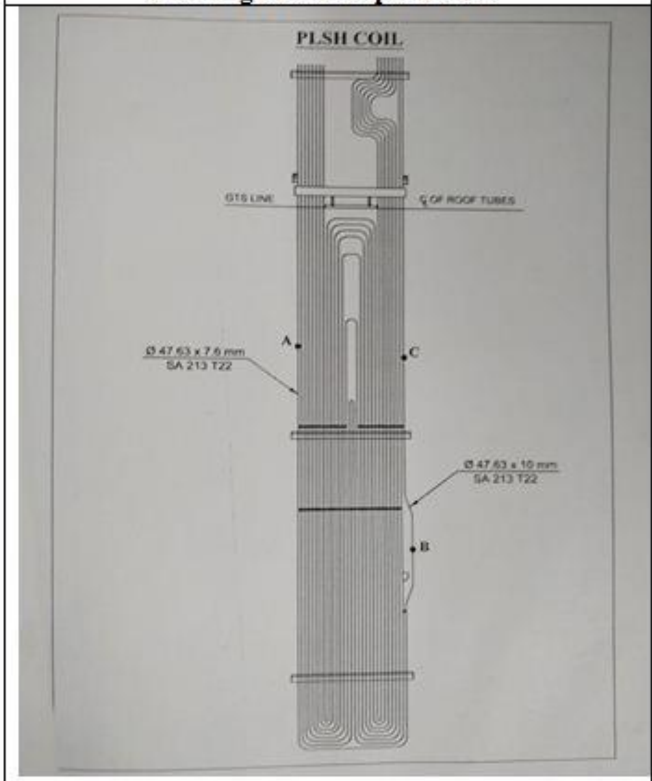
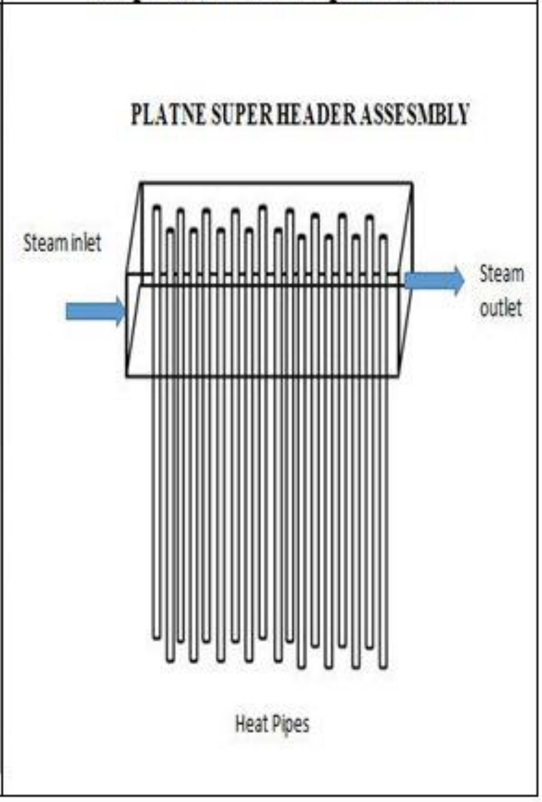
Existing Platen Super heater	Proposed Platen Super heater
	

Fig. 4.12. Proposed assembly of Platen Super heater

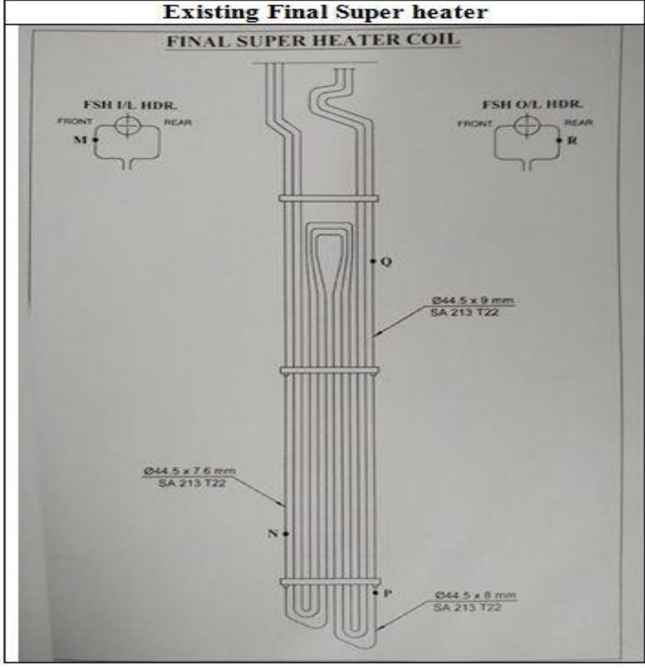
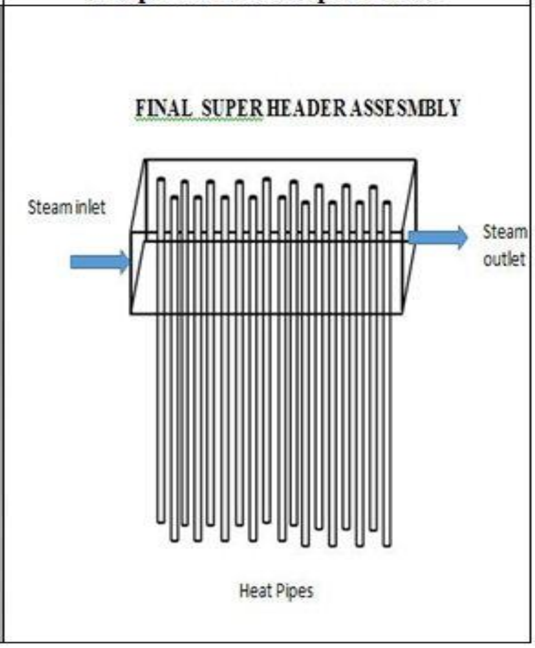
Existing Final Super heater	Proposed Final Super heater
	

Fig. 4.13. Proposed assembly of Final Super Heater

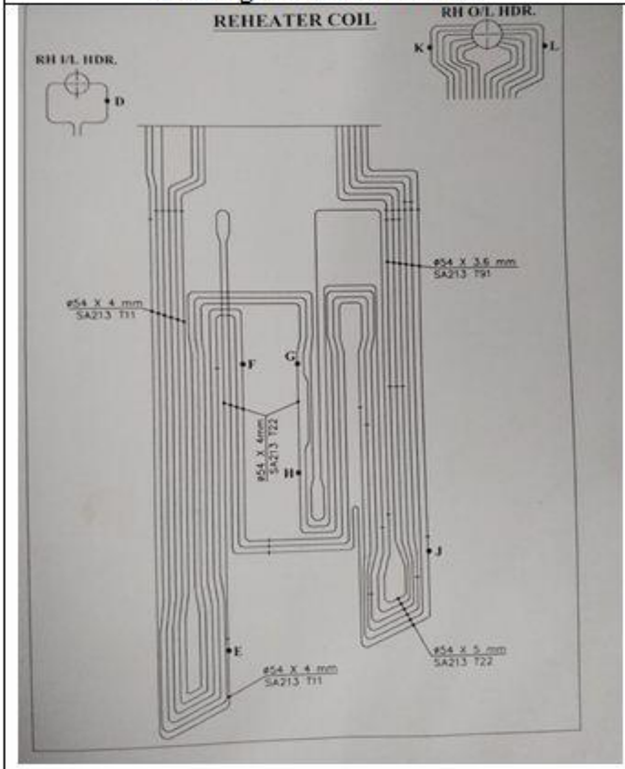
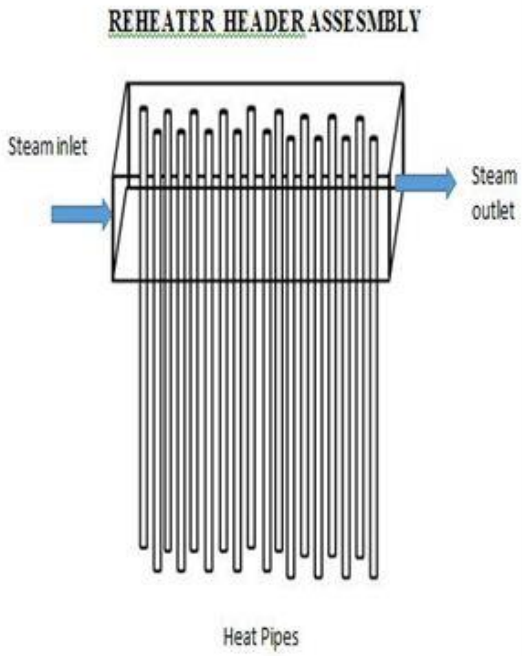
Existing Re heater Header	Proposed Re heater Header
	

Fig. 4.14. Proposed assembly of Re heater

The table top model of proposed heat pipe based super heater and its rectangular header shown in Fig 4.15

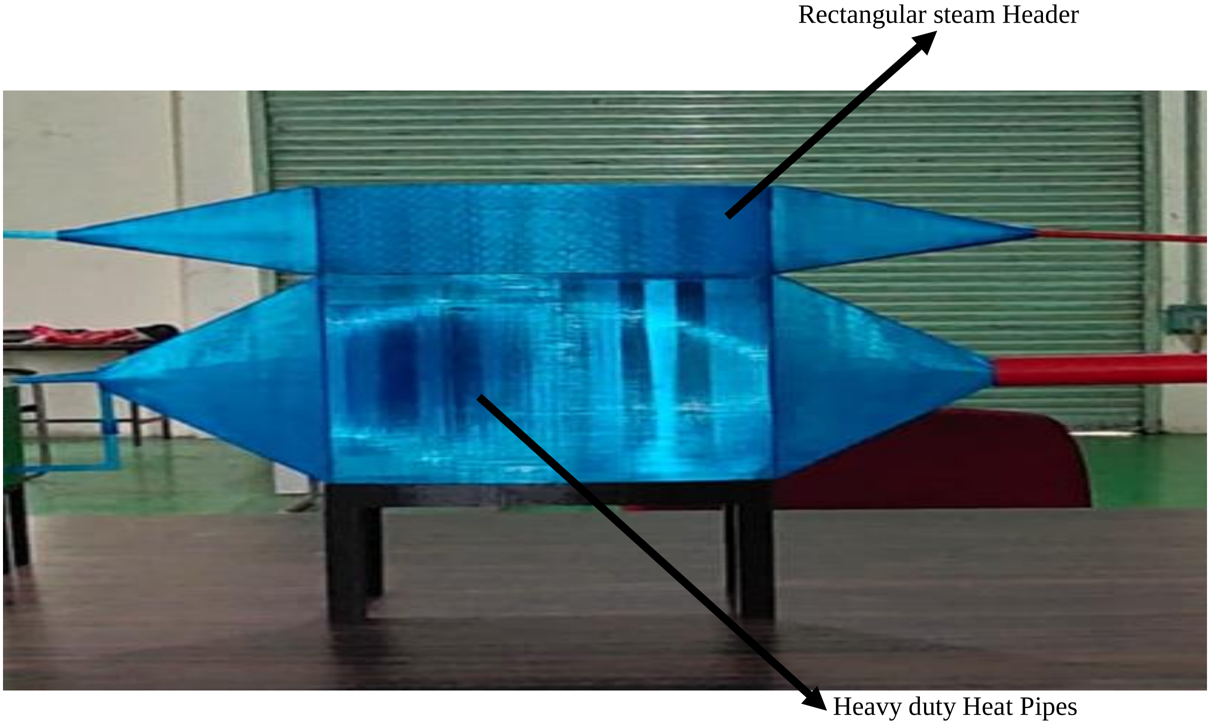


Fig. 4.15 Table Top Model of the proposed heat pipe super heater & header

With this arragment the heat transfer efficiency will increase and also the exergy desturction will decrease as shown in Table 4.7 (The detailed calculations reported in Appendix VIII) in the Platen super heater,Final super heater and re heater assemblies.

TABLE 4.7. EFFECT OF EXERGY DESTURCTION DETAILS WITH USE OF HEAT PIPES

Component	**Exergy destrction in MW**		**Improvement in Exergy destruction**
	Conventional,, as existing now	With proposed modifications	
Platen Super heater	30.4	10.4	65.8 %
Final Super heater	14.9	5.58	62.5 %
Reheater	30.9	10.7	65.4 %

Due to above modifications the overall boiler exergetic efficency will increase to 59.0 % from the 52.9 %. Also with this heat pipe super heaters system , the floor space in the penthouse of the boiler will be reduced significantly. Because of the reduced prssure drop across these proposed heat pipe assemblies , the load on the boiler feed pump reduces which again leads to the improved the exergetic efficieny of the boiler.

The exergetic efficiency of boiler with heat pipes is compared with exergetic efficiencies of other power plant boiler available in the published data (from the references) and presented in Fig 4.16.

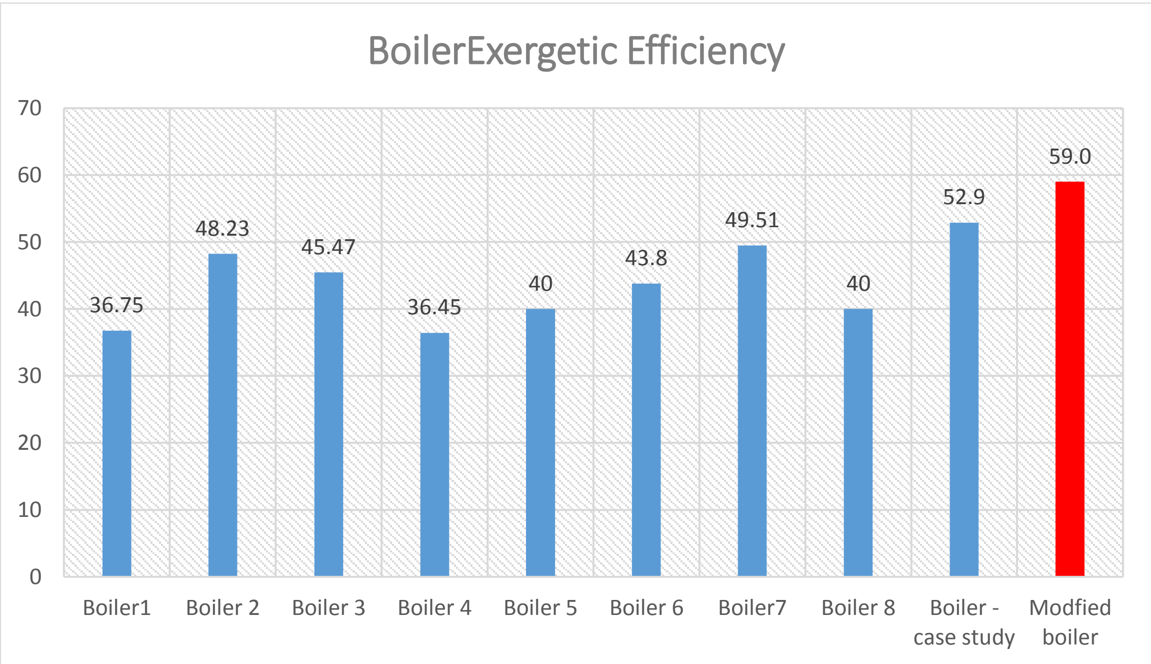


Fig 4.16. Comparison of Exergy efficiency of boilers with modified boiler

4.2.4 *Proposal of Technology to improve 5E parameters and reduce the Carbon footprint in the Plant by Steam blending with Solar Tower Technology*

In the conventional steam power plant, the water/steam circuit will be as shown in Fig 4.17. The condensate water along with the makeup water (from water source) will be pumped into the economizer. In the economizer the pre heated feed water, raises to a temperature of 300 ^{o}C and then being fed into the boiler. The boiler which consists of super heaters raises the temperature of steam up to 540 ^{o}C which in turn will be fed into the turbine for conversion of thermal energy into mechanical energy to rotate the generator.

For the considered case study plant, and for the given output, the following modification will improve the efficiency and exergy efficiency of the plant. The modifications use the solar heating and coal burning hybridization as shown in Fig 4.18.

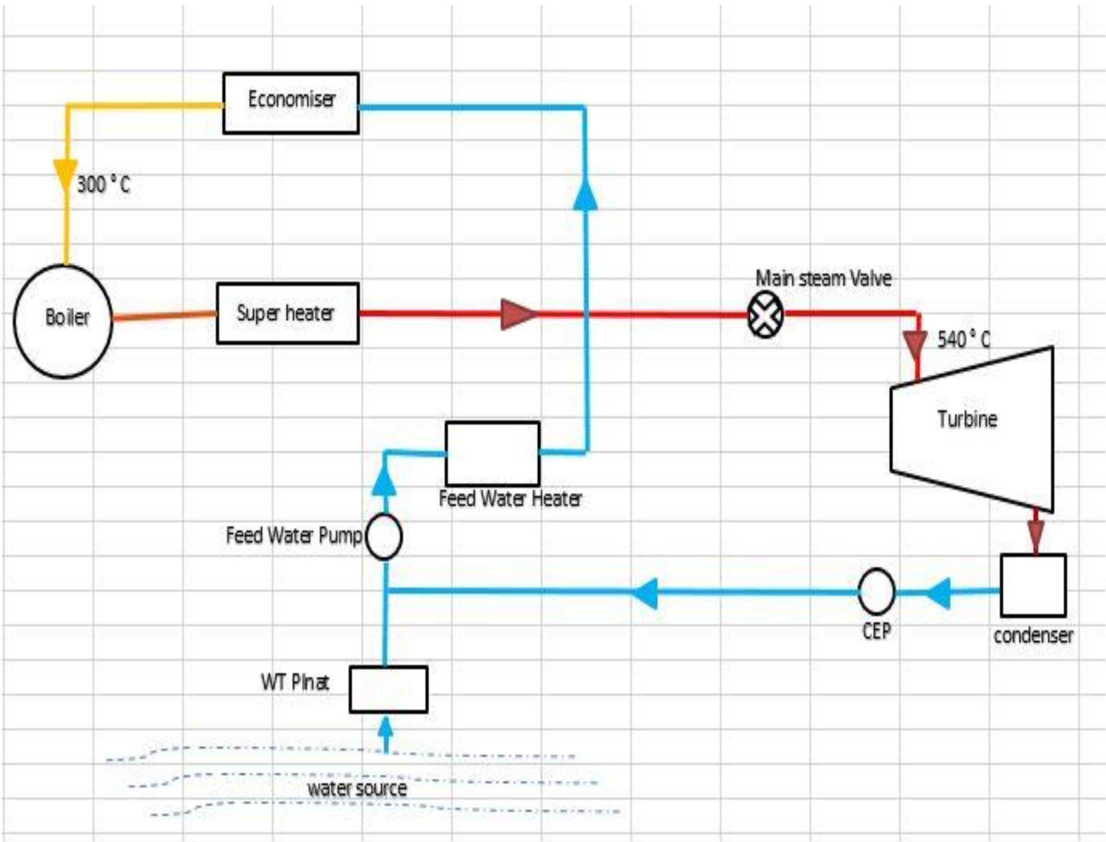


Fig 4.17: Water /Steam circuit of Steam Power Plant

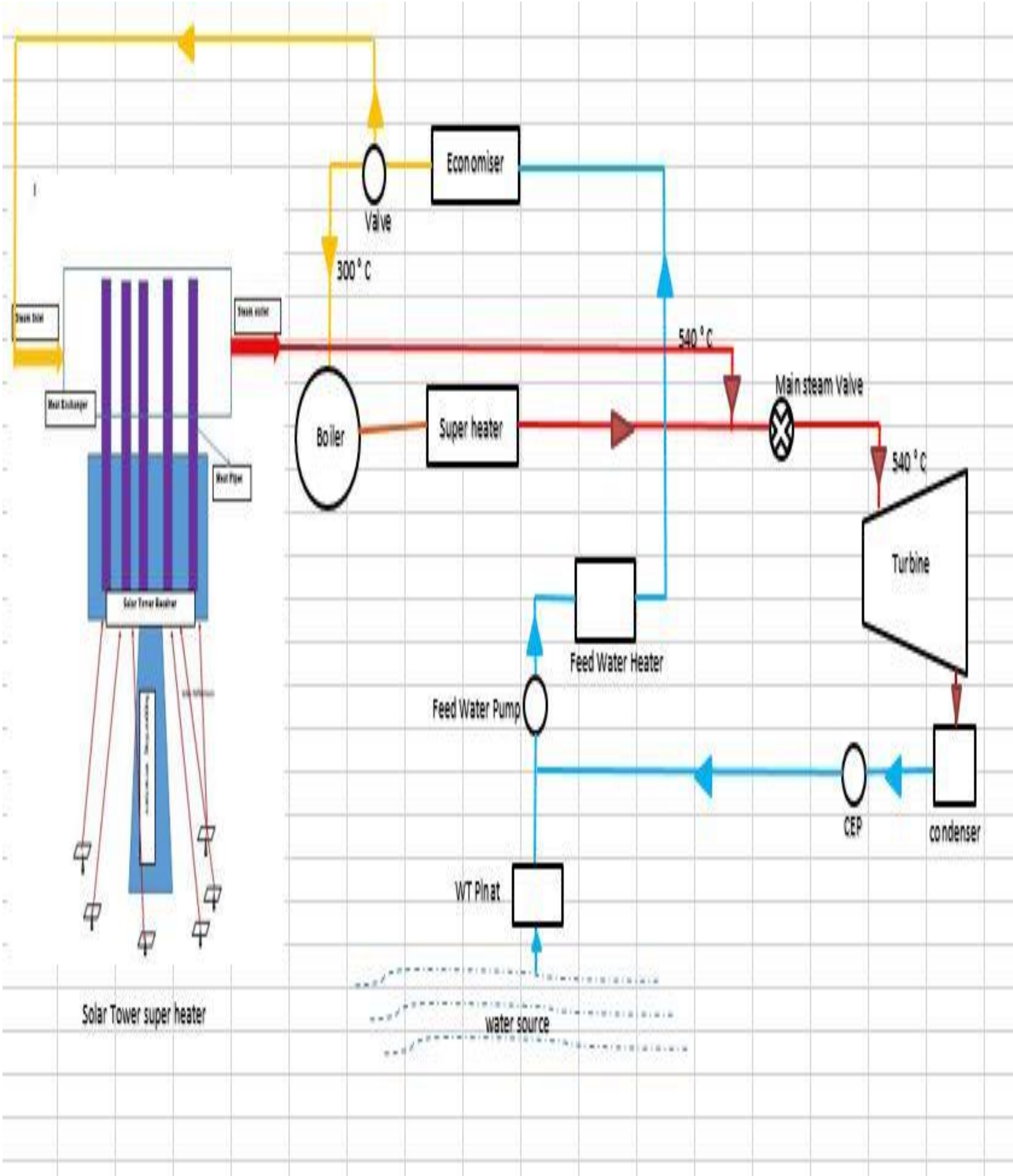


Fig 4.18: Modified water/steam circuit of proposed steam power plant

The economizer outlet will be connected to the inlet of solar heating system. In this solar heater, the economizer outlet steam which is at a temperature 300° C will be heated to a temperature of the 550° C. Now this super-heated steam is blended with the steam produced from the coal. Hence, the coal consumption will be reduced, efficiency increases and exergy destruction reduces and also the emission of CO_2 will be reduced. Because of this blending of steam, the output of steam power plant will not be affected.

The solar steam blending can used from 8.30 AM to 4.30 PM (In India). During this period, 60 % of steam that is 332.64 t/hr will be produced by solar and remaining 40 % of steam that is 221.76 t/hr of steam will be produced by coal burning

With above proposal, the case study plant, which considered in this work, for 210 MW output, the parameters will improve as in Table 4.8.

TABLE 4.8. COMPARISON THE PARAMETERS WITH PROPOSED SOLAR STEAM BLENDING

Sl. No	Description of Parameter	Existing Plant	Modification with Solar Blending	% Improvement
1	Coal quantity	40 kg/s or 3456tons per day	32 kg/s or 2764.8 tons per day	20 % coal saving
2	Coal quantity per day	3456 tons	2764.8 tons	691.2 tons of coal saving
3	Carbon Di oxide (CO_2) emission per day	3294.5 t	2635.8	19 % of CO_2 that is 658.7 tons of Co_2 reduction
4	Exergy destruction in the boiler	400 MW	366.61 MW	8.4 % saving in the exergy destruction

A. *After modifications with Heat pipes and Solar steam blending, the efficiency and exergy efficiency will be,*

Efficiency (η)

η = Energy output / energy input

= (Energy out – Auxiliary consumption) / (Calorific value of coal x coal consumption + Solar Energy input)

= 32.2 %

Exergy efficiency ($Ɛ_\eta$)

For the typical Indian conditions and geographical location of the case study plant, solar energy input [58,59] with solar heliostats and central receiver

Solar Energy received = 97,740 kW

And exergy input due to solar exergy = 90,810 kW

Total exergy input into boiler by coal and solar = 816.71 MW

Exergy out from the boiler = 450.1 MW

Exergy destruction in the boiler = (816.71-450.1) = 366.61

$\exists_{\eta,b}$ = Exergy output from boilr/Exergy input to boiler

= 55 %

Proposed method of Using Solar Tower Technology for super heating the Steam is presented in the Appendix IX.

4.3 Propsals to dcreae Exergy destruction in Turbine

The line digram of existing High pressure turbine, itnermediate pressure turbine and Low pressure turbine shown inf Fig 4.19

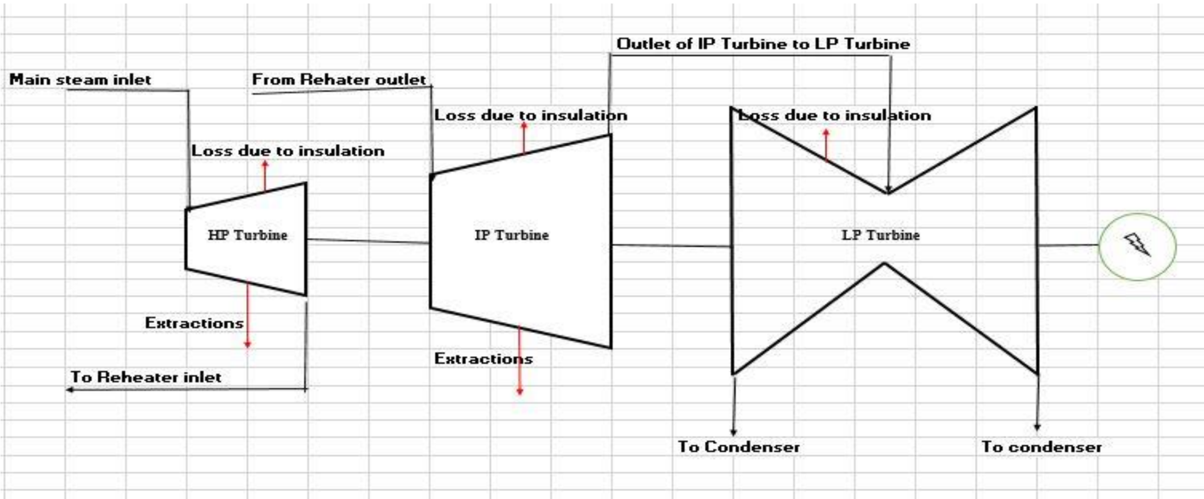


Fig 4.19. Line digram of Turbine Arrangment

There are three separate barrels in which HP rotor, IP rotor and LP rotor are encapsulated with insulation.

Exergy Calculation Existing Turbine (The calculations are presented Appendix II)

- Exergy Destruction of HP turbine = 8.85 MW
- Exergy Destruction of IP turbine = 0.3 MW
- Exergy Destruction of LP turbine = 7.0 MW
- Overall Turbine Exergetic Efficiency = 97.0 %

. *4.3.1 Probable Reasons for Exergy Destruction*

Irreversible nature of steam expansion in blades

Exhaust steam losses

Heat Transfer between the high temperature turbine

barrels and atmosphere. This is because of relatively poor insulation on turbines hoods.

4.3.2 Proposal for decrese the exergy destruction

The present insulation on turbine barrels is designed to maintain surface temperature of 60^{o} C. It is proposed to improve the insulation properties so that surface temperature will maintained at 40^{o} C , so that exergy destruction due to heat loss will be reduced and exergetic efficieny will improve by 0.1 % . (Detailed calucation are presented in Appendix VI)

The exergetic efficiency Turbine with proposed modification is compared with exergetic efficiencies of other turbines available in the published data and presented in Fig 4.20.

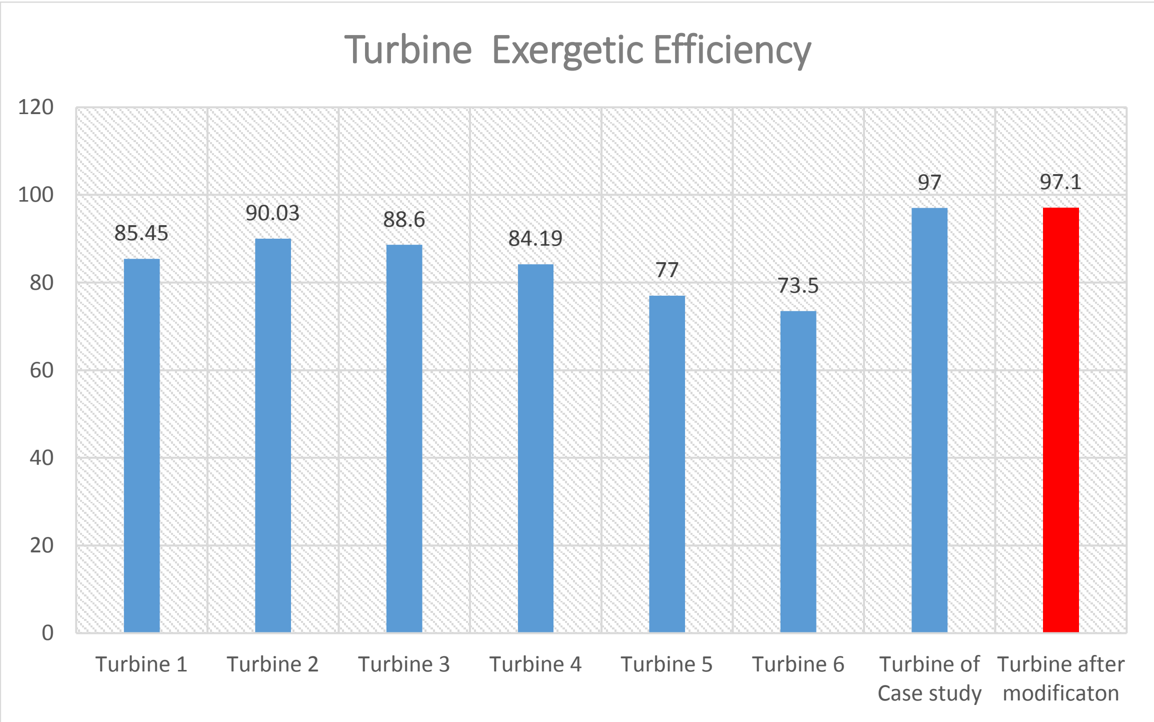


Fig 4.20. Comparison of turbine Exergetic Efficiencies with modified turbine

4.4. Conclusions for this Chapter:

The summary of the above all proposals can be summarized as in Table 4.9. The 5E calculations after proposed modifications are presented in Appendix IX.

TABLE 4.9 EFFECT OF MODIFICATIONS ON 5E PARAMETERS

E description	Existing Plant	Modifications with heat pipes	Modifications with heat pipes and Steam blending with solar tower
1E (efficiency)	31.09%	31.12 %	32.2%
2E (exergy efficiency)	49.7%	53.5 %	52.4 % (Carbon footprint reduced by 19%)
3E (Exegoeconomic factor)	417.2	616.2	927.3
4E (Exergoenvironmental index)	18 %	19.1 %	25.4 %
5E (Endurability Factor)	0.689	0.599	0.581

Thus with the above suggested proposals the 5E parameters will improve and thermal power plants become more viable from environmental point of view and also carbon footprint will be reduced considerably. Thus sustainability of power plants will improve. The solar blending suggested can be implemented on the existing power plants, with the available space around plant.

****

Chapter 5	Effect of Proposals and Discussion

5.0 Introduction

In the previous chapter number of reforms are suggested for improving the viability of the thermal power plants. In this chapter the results of these reforms will be discussed.

5.1. Condenser

In the condenser, the heat pipes are suggested as replacement for nonferrous tubes in the condenser.

Advantages:

1. The exergy destruction in the condenser will reduce and hence the condenser exergetic efficiency may increase up to 39%. Hence the thermal pollution caused by condenser will reduce significantly.
2. Use of heat pipes will reduce the usage of number of nonferrous tubes in the condenser for the given load. Due this the maintenance will be easy and also maintenance cost will reduce.
3. The size of condenser will be reduced significantly.

5.2. Boiler

The re designing of heat transfer components of platen super heater, final super heater and re heater are proposed with heat pipes. In addition, the steam blending with solar heated steam is also suggested.

Advantages :

1. The redesigning of super heaters and re heaters with heat pipe will improve the exergetic efficiency of boiler by 4%.
2. The number of boiler tubes will be reduced hence less maintenance and also space will be saved.
3. Boiler forced shut down will be avoided because tube failures in the super heater and re heater section, which is a greater advantage.

In addition to heat pipe super heaters and re heaters the solar steam blending also proposed.

4. This blending will improve the boiler exergetic efficiency by 21 %.
5. Coal consumption will be reduced by 30 % without reduction in the output.
6. The Co_2 and other pollutant gas emissions will be reduced.

5.3 Turbine

The improvement of insulation properties on the turbine barrel suggested.

Advantages:

1. The exergy destruction will reduce to an extend of 0.1 %
2. The thermal pollution will be reduced.

5.4 Overall Advantages:

The usage of Heat pipes in the super heater, platen super heater, re heater and condenser will improve the 5E parameters with a significant improvement. Not only, efficiencies but also exergoceonomic factor and endurability factors also improves significantly**.** All the modifications and designs are proposed, keeping the space and design constrains of the existing plant so that the proposals can be adopted in the existing plants also.

If space and sources are available, then solar steam blending can also be implemented which improved endurability factor of power plant to a notable extent.

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Chapter 6	**Experiments with Heat Pipes**

6.0 Introduction

To improve the competency and gentleness of the thermal power plants to wards environment, different technologies have been suggested in the chapter 4. Almost all technologies are based on the usage of heat pipes like heat pipe based condenser, heat pipe based super heaters & re heaters and also heat pipe heat exchanger for solar steam blending. Heat pipes are one of the most versatile man-made heat transfer device whose application would improve the sustainability of the power plant. Further to prove the application heat pipes to power industry, in this work, heat pipes are designed and developed for steam condensation. Laboratory experiments were carried on single heat pipe and also with bunch of heat pipes in laboratory model condenser. This validates one of proposed technology of steam condensation with heat pipes.

6.1 Design of Heat Pipe

In these experiments, the heat pipe was designed to condense the steam at a temperature of 46˚C and cooling water at a temperature of 26 ˚ C. The steam parameters chosen were similar to the turbine exhaust conditions of steam in a steam power plant. The pressure inside the heat pipe is fixed such that the working fluid inside the heat pipe absorbs the heat energy from the steam surrounds the evaporator section of the heat pipe and gets converted into vapour. Then, the vapour travels to the condenser section of heat pipe and releases the absorbed heat energy to a cooling fluid present around the condenser section of the heat pipe. The details of the designed heat pipe are presented 4.1.2**.** The Heat pipes with the above design details were fabricated by M/s. Thermosys heat pipe manufacturers, Vadodara, Gujarat – 390 010, India. The photograph of the fab fabricated heat pipe is shown in Fig 6.1

Fig 6.1 Fabricated Heat Pipes

6.2 Designing and fabricating the steam generator/Boiler required for the condensation experiments

The steam generator required for the experiment was designed and fabricated in the laboratory. The line diagram of steam generator and actual photograph is shown in Fig 6.3. The boiler is fitted with 4 x 3 kW heaters and also 3 x 9 kW heaters so that energy input can be varied according to requirement. All the heaters are electrically controlled. The boiler is fitted with a fine mesh screen so that the liquid droplet will be screened down. The Fitting of heaters into the condenser also shown in Fig 6.2. The heating of boiler for the supply of steam is shown in Fig 6.3

6.3 Experiments and Discussion of Results

The performance of the designed heat pipe and its capability to condense the steam is evaluated in 3 steps, viz:

A. Experiments on a single heat pipe with hot water.
B. Experiments on a single heat pipe with steam.
C. Experiments on the condenser loaded by stock of heat pipes with steam.

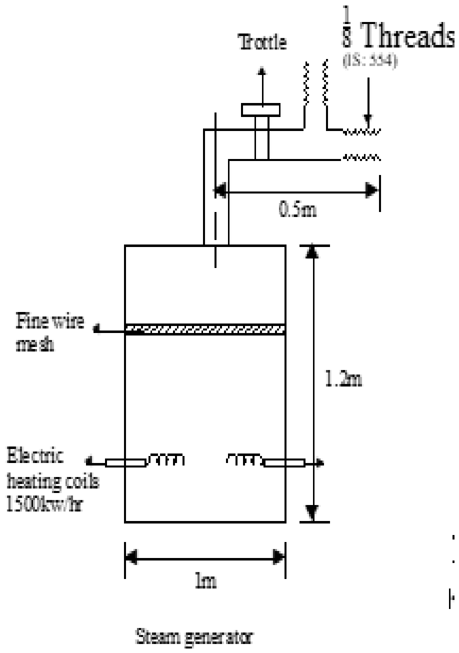


Fig 6.2. Steam Generator and accessories at Different Stages of Fabrication

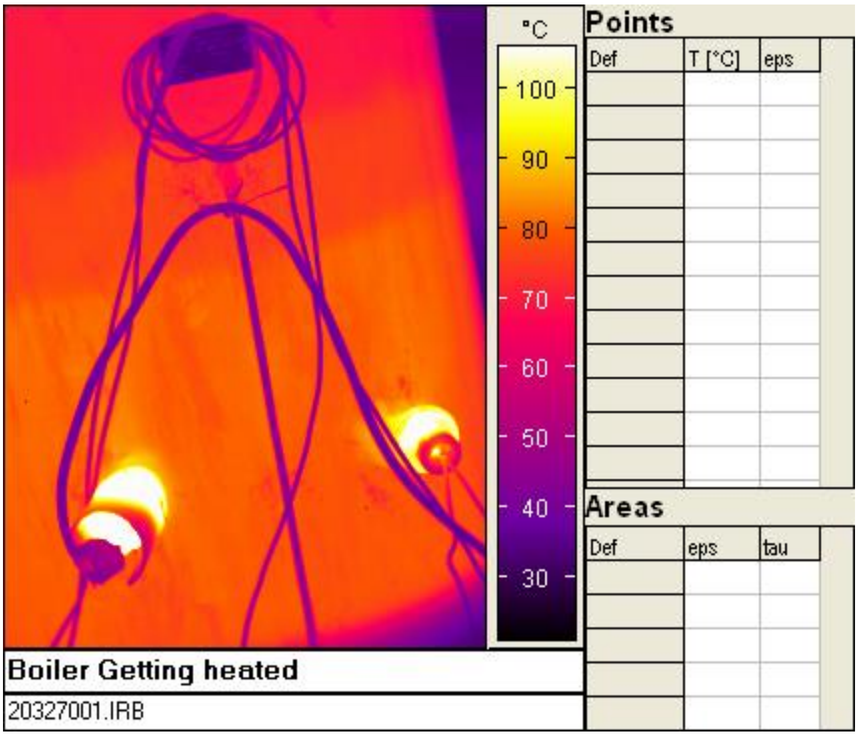


Fig 6.3 The thermogram of boiler heating

6.3.1 Experiments on s single heat pipe with hot water.

The line diagram of the heat pipe and actual photograph of the experimental set up is presented in Fig 6.4. The evaporator section of the heat pipe was enclosed with a jacket (made of PVC pipe) which acts like hot water enclosure. This enclosure is provided with two openings, one for hot water inlet and the other for hot water outlet. The temperatures of hot water at inlet and outlet of the jacket were measured with the help of digital thermometers. The condenser portion of the heat pipe was enclosed with another jacket which acts like a cooling water enclosure. This enclosure was also having two openings, one for cooling water inlet and the other for cooling water outlet. The inlet and outlet temperatures of the cooling water were measured with digital thermometers at these openings. The middle portion of the heat pipe was insulated.

The portion of the heat pipe enclosed with hot water jacket acts as the evaporator section of the heat pipe and the portion which is enclosed by cold water jacket acts as condenser section of the heat pipe. The middle portion of the heat pipe is insulated and it acts like the adiabatic section.

The single line diagram of the experimental set up For Single Heat Pipe

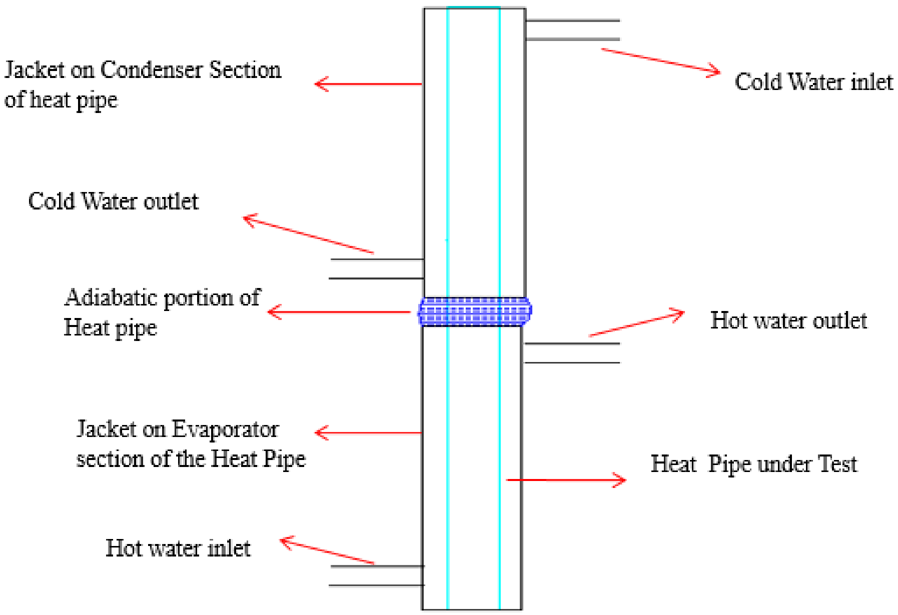


Fig 6.4. Line Diagram and Experimental Set up for Experiments with hot water

Experiments

Hot water is allowed to enter the hot water enclosure from bottom opening and this water will exit from the top opening of the hot water enclosure. Cold water is allowed to enter the cold water enclosure bottom opening and allowed to exit from the top opening. The inlet and outlet temperatures are measured with the digital thermometers placed at the inlet and outlet openings. Flow meters were fixed in the hot and cold water line to measure the flow rates of hot and cold water supplied to the heat pipe. The readings are noted after steady state was attained. Number of trials were conducted on this experimental set up using hot water at different temperatures and the energy transported by the heat pipe was determined and presented in Table A.5 of Appendix XI. The performance of the heat pipe with hot water experiments was presented in the Fig 6.5.

Heat Balance, for a typical reading,

Flow rate of hot water = 6.19 lit/min = 0.103 kg /sec ,

T_1 = Hot water inlet temperature = 53.26 ^{0} C

T_2 = Hot water outlet temperature = 42.49 ^{0} C,

Heat energy supplied to the heat pipe = 0.103 x 4.180 x (53.26-42.49) = 4.64 kJ/sec

This is amount of energy transported by heat pipe to the cold fluid.

Cold water flow rate = 6.56 lit/min = 0.109 kg/sec,

T_3 = Cooling water inlet temperature = 25.79 ^{0} C

T_4 = Cooling water outlet temperature = 35.55 ^{0} C

Heat energy gained by cooling water = 0.109 x 4.180 x (9.76) = 4.45 kJ/sec

The above experiment indicates that the heat energy supplied to heat pipe is being transported to the cold fluid with an efficiency of 96 %. Hence it can be concluded that the designed heat pipe is working satisfactorily.

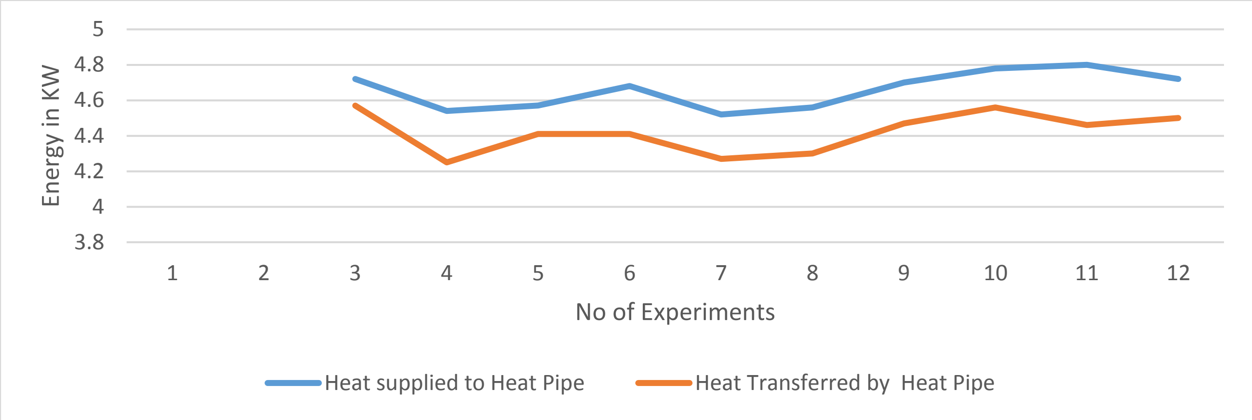
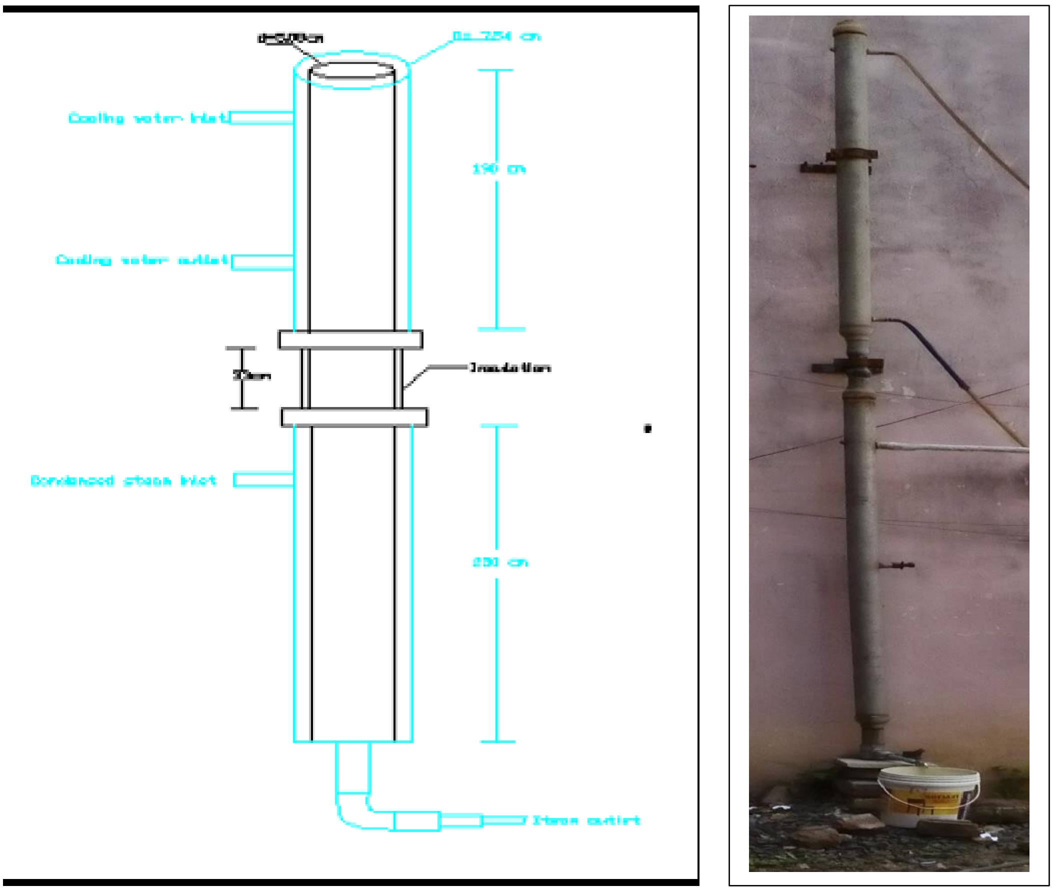


Fig 6.5. Performance of heat pipe with Hot water

Discussion of the Results. The above experiments and calculations prove that the designed heat pipe can transport the heat energy successfully. The performance is consistent over the entire range of experimental values that is 42^o to 66.4^o C. Hence, it is decided to two conduct two phase experiments on this heat pipe to test the capability of the heat pipe for steam condensation.

6.3.2. Experiments on a single heat pipe with steam.

The line diagram and actual photograph of the experimental setup is presented in Fig 6.6. The heat pipe lower portion was enclosed with a jacket of GI pipe in which steam was injected for condensation purposes. Top portion was enclosed with another jacket which acts like a cooling water enclosure. The portion of heat pipe enclosed for steam condensation acts as the evaporator of the heat pipe and the portion which was enclosed by cold water jacket acts as condenser of the heat pipe. The inlet and out let temperatures were measured with the digital thermometers placed at the inlet and outlet openings.



(a) (b)

Fig 6.6. The line diagram and actual photograph of the experimental set up for experiments with hot water

Steam with different temperatures was allowed to enter the steam enclosure from the inlet opening and allowed for condensation. The condensate will be flow down by the tap. Cooling water flow form bottom opening to top opening of cold water enclosure. The temperatures are measured at respective passages. The energy supplied through heaters is 39 kW. The steady state readings are presented in Table A.6 of Appendix XI. The flow of Condensate is shown in Fig 6.7. Repeated experiments were conducted on this experimental set up using steam at different temperatures and energy transported by the heat pipe was determined. The performance of the heat pipe with steam was presented in the Fig 6.8.

Fig 6.7. The flow of Condensate as a result of Steam Condensation

Heat Balance, considering a typical reading,

Q_1 = Heat input of condensate in kW = 29.659

(Current drawn from mains = 139.8 amps, voltage = 221V, p .f is 0.96)

Q_2 = Heat energy carried out by condensate in kW

= mass of condensate m/s x Enthalpy of condensate

= 0.0126 x 184.3 = 2.32

Q_3 = Heat output of cooling water in kW

= mass of cooling water x specific heat x temp. rise

= (33/60) x 4.180 x (32-22.5) = 21.84

$\Delta Q = (Q_1 – Q_2) – (Q_3) = 29.659 – 2.32-21.84 = 5.5$ kW

Discussion of Results. The above experiments and calculations prove that the designed heat pipe can condense steam successfully. The performance is consistent over the entire range of experimental values. Hence it is decided to conduct experiments on the stock of heat pipes in a condenser to test the capability of heat pipes for steam condensation. The above results indicates that designed heat pipe working satisfactorily with steam also.

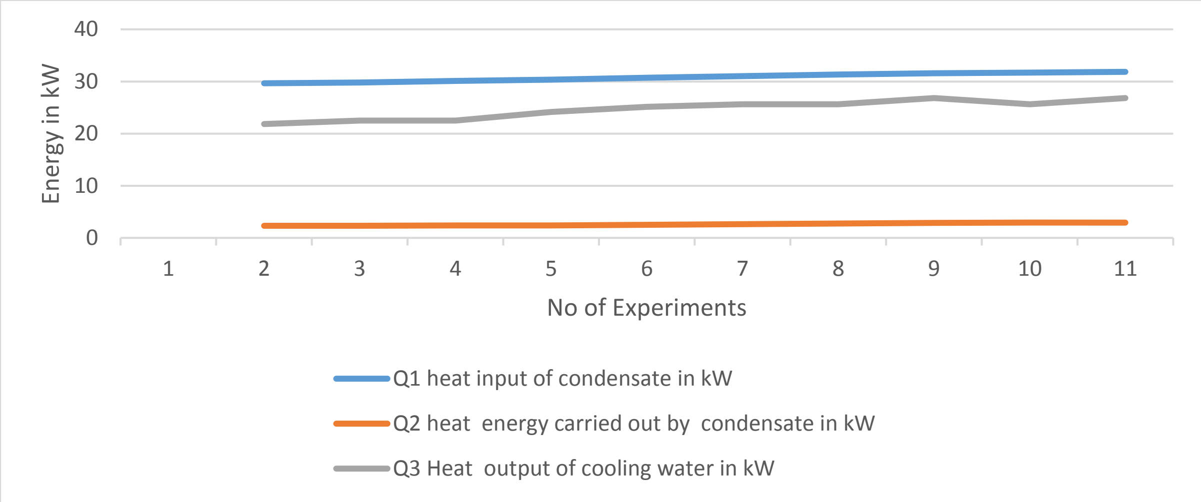


Fig 6.8. Performance of the heat pipe with steam

6.3.3 *Experiments of condenser loaded by heat pipes with steam.*

The designed heat pipe successfully working with hot water and steam. Hence it is decided to conduct experiments on the bunch of heat pipes to test the steam condensation capability.

Experimental Set up & fabrication of the heat pipe condenser

The heat pipe condenser required for the experiment was designed and the schmatic diagram of the steam condenser experimental setup, table top model and the actual photographs are shown in Fig 6.9, 6.10 and Fig 6.11.

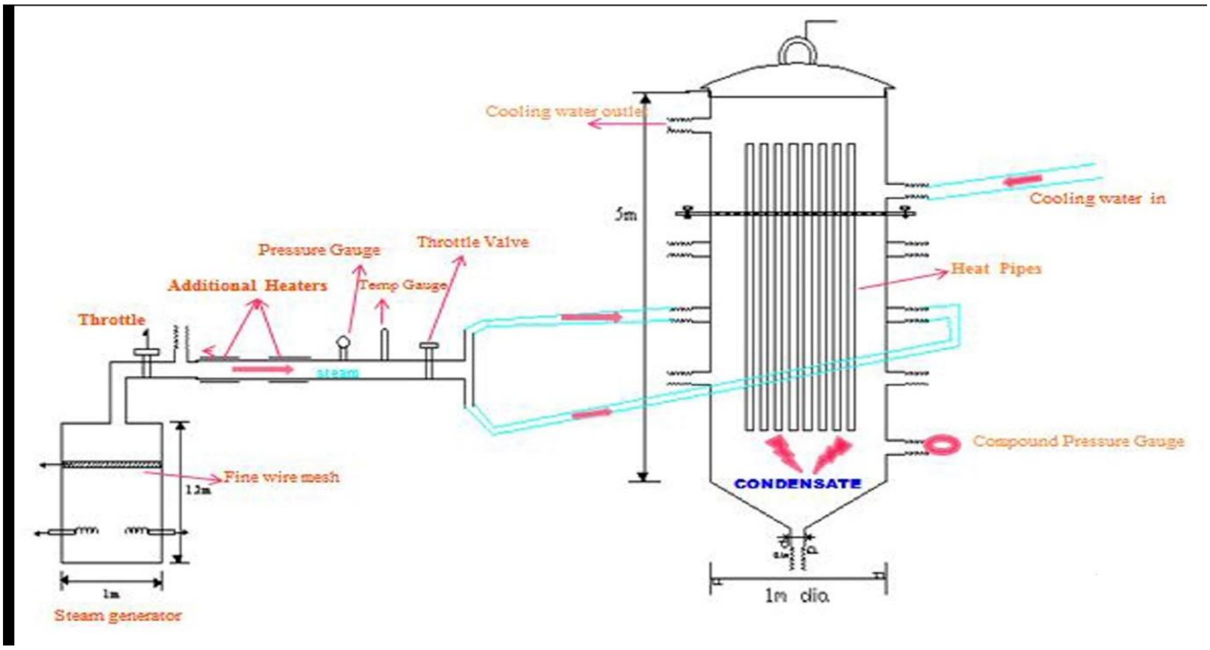


Fig 6.9 schematic drawing of Heat pipe condenser experimental set up for heat pipe condenser experiments

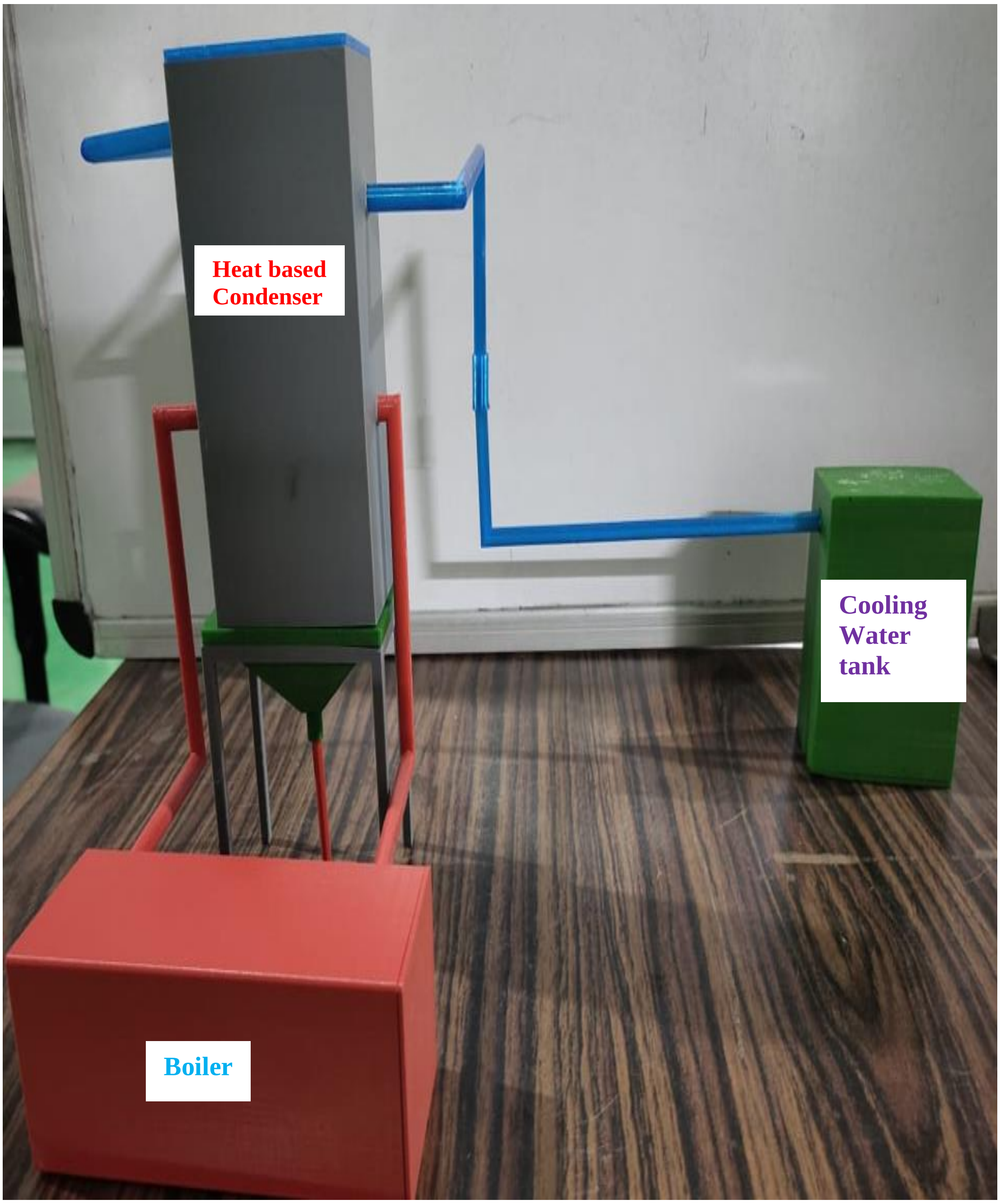


Fig 6.10 Table top model of the Experimental Set up for heat pipe condenser epxeriments

Instruments Used during Experiment: Flow Meters, Digital temperature indicators, compound pressure gauge, vacuum pump. Bucket and beaker (to measure the quantity of cooling water) Power Analyzers to measure the power input to boiler heaters.

Description of the Experiment: The condenser is separated into two halves with a separator. The bottom portion, which acts as evaporator has steam entry and condensate outlet. The top portion which acts a condenser has cooling water entry and exit.
No of heat pipes used are 16. The steam being supplied by a steam generator of capacity 39.0 kW. A superheating system is arranged to superheat the generated steam. Then the superheated steam is throttled and fed into the condenser. The temperature and pressure of the steam were measured before the throttle valve. The actual photo of the experimental set up is given above. After reaching the steady state conditions, the readings were taken and presented in Table A.7 of Appendix XI and the performance of heat pipe based condenser is shown in fig 6.12

Model Calculations:

Q_1 = Heat input of condensate in kW = 30.734

(Current drawn from mains = 145.5 amps, voltage = 220V, p. f is 0.96)

Q_2 = Heat energy carried out by condensate in kW

= mass of condensate $\dot{M}$ per second x Enthalpy of condensate

= (0.85/60) x 191.8

= 2.72

Q_3 = Heat output of cooling water in kW

= mass of cooling water x specific heat x temp. rise

= (35/60) x 4.180 x (36.3-26.0)

= 25.12

$\Delta Q = (Q_1 - Q_2) - (Q_3) = 30.734 - 2.72 - 25.12 = 2.89$ kW

The performance of the heat pipe based condenser experiments cane be represented in the following Figure.

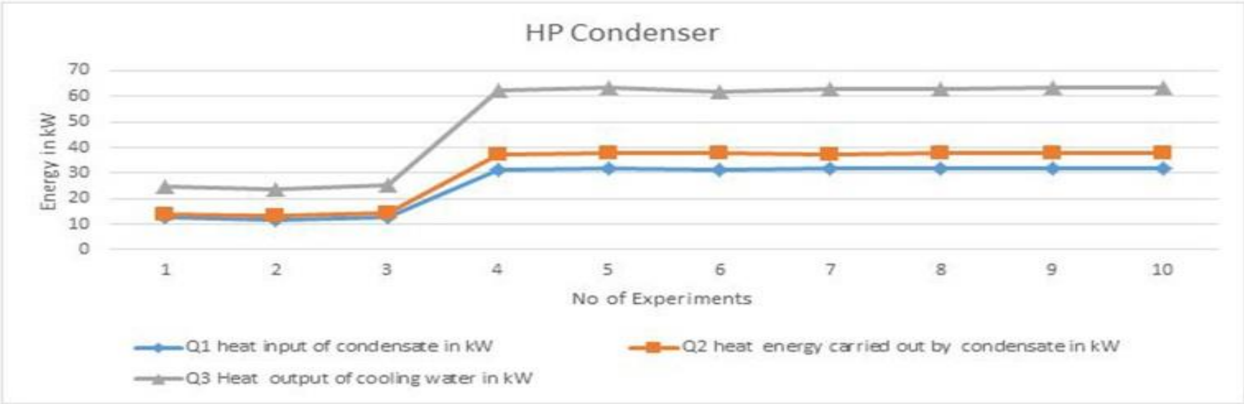


Fig 6.12 Performance of heat pipe based condenser

Discussion of Results. The above experiments and calculations prove that the designed heat pipe in stock can condense steam successfully. The performance is consistent over the entire range of experimental values. Hence it can be concluded that the designed heat pipes in stock are working satisfactorily for condensation purpose.

6.4 Uncertainty Analysis

Uncertainty analysis for the experiment's equipment was carried out based on the reference [50,51]. The results and used formulae are presented in Appendix XII. The inaccuracy of the measured parameters are below 10%. Hence it can be safely concluded that the results obtained are reliable and repeatable.

6.5 Conclusions for this Chapter

This heat pipe was experimented with hot water and steam and got encourage able results. Then a laboratory model of heat pipe based condenser was fabricated and experimented were carried out with steam. The heat pipe condenser worked very well and successfully condensed the input steam.

Thus one of technology to improve the 5E parameters of the steam power plant got validated.

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Chapter 7	Final Conclusions

7.0 Conclusions

The thermal power generation cannot be eliminated in the near future, because of uncertainties in the renewable source power generation. To keep the survey ability of thermal power station, the performance of thermal plants are to be improved and also the ill impact of thermal plants on the environment has be minimized. Also the now non-renewable fuels like coal and oil are to be consumed in optimum way. Significant conclusions of this work are,

7.1 Plant Evaluation by 5E

The power plants are to be evaluated more efficaciously by 5E parameters that is energy, exergy, exergoeconomic, exergo environmental and endurability analysis. The exergoecomic factor tells the economic implications and exergo environmental effects of plant on the nature. The endurability factor informs the sustainability of the energy resources.

7.2 Design Modifications

Different design modifications and process modifications are suggested. As result of these modifications

- 1E (efficiency) improves from 31.09 % to 32.2%
- 2E (exergy efficiency) improves from 49.7 % to 53.5 %
- 3E (exergyeconomic factor) improves from 417.3 to 927.3
- 4E (exergo environmental index) improves from 18 % to 25.4 %
- 5E (endurability factor) improves from 0.689 to 0.581

7.3 Usage of Heat Pipes in Boiler

The usage of heat pipes in the heat transfer equipment of boiler will have following effects

- The exergy destruction reduction improvement in Platen super Heater will be 65.8 %, Final super heater will be 62.5 % and re Heater will be 65.4%

- Reduction in the heat transfer area (around 70%), which decreases numbers tubes in the heat transfer equipment and hence less maintenance problems
- Boiler Tube Failures will be reduced, hence Forced Shutdowns may be reduced.

7.4 Solar Blending

The steam blending with solar heated steam as suggested , improves

- Reduction in Coal consumption by 20%
- Reduction in Co_2 production will be 19%

7.5 Usage of Heat Pipes in Condenser

The usage of heat pipes in the condenser will have following effects

- The exergy efficiencywill improve to 61% from 22%
- The effectivensees of heat transfer rate improves due to which heat area will decrease, hence there will be 53 % reduction in number of heat transfer tubes.
- Ease of maintenace of condenser.

7.6 Improving the Insulation Properties

- The suggestion of improvement of turbine hood insulation properties improves the exergetic efficiency by 0.1 %

7.7 Future Scope of Work

The viability of the TPP can be further improved by reducing exergy destruction in boiler (by improving combustion, augmenting feed water heaters, heat transfer mechanism of boiler water wall tubes etc.), turbine (reducing the irreversible nature of steam expansion in the blades).

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Publications: Journal Papers

S.No.	Title	Name of the journal and publisher	Status
1.	Analysis of Power Plant more efficaciously and reforms by Heat Pipes	International Journal of Ambient Energy, Taylor & Francis, (SJR ,ScopusJournal)	Published Vol 43,No 1,8067-8079 2022 Doi.org/ 10.1080/01430750.2022.2085794
2.	Modifications of Sub-Components in Thermal Power Plants for Exergetic Efficiency	International Journal of Heat and Technology , IIETA (SJR,Scopus)	Published Vol 39,No.2,573-580 April 2022 doi.org/10.18280/ijht.390227
3.	Steam Condensation by Heat pipes	International Journal of Physics" (Scopus indexed),	Published 1473(2020)012028 doi: 10.1088/1742-6596/1473/1/012028
4.	Comparison between Heat Pipes Based Condenser and Conventional Condenser of Power Plant ,	World Journal of Physics, Science Publishing Group	Published 2022:7(3):33-42 doi: 10.11648/j.wjap.20220703.11

Publications: Conferences

S. No.	Title	Name of the conference	Status
1.	Retrofitting of power plant condensers with heat pipe heat exchanger " A case study"	VIII Minsk International Seminar,"Heat Pipes, Heat Pumps, Refrigerators, Power Sources", Minsk, Belarus,	Presented and Published in the seminar proceedings
2.	Steam Condensation by Heat Pipes	Phase change thermal systems' at IIT, Kanpur	Presented

Annexures

Appendix I

Case Study Power Plant Parameters

TABLE A.1. OPERATING PARAMETES OF BOILER

Sl.No	Parameter	Numerical Value
1	Plant Name plate capacity	210 MW (191 MW during time of consideration)
2	Boiler	Π (pi) type, pulverized coal fired boiler
3	Calorific value of coal	14,654.5 kJ/kg
4	Quantity of coal used	40 kg/s
6	Mass of primary air	281 t/hr at temperature of 325℃
7	Mass of secondary air	655 t/hr at temperature of 318℃
8	FD Fans rating (2 Nos)	750 kW
9	PA Fans rating (2 Nos)	1250 kW
10	Power Required for Boiler Feed Pump (2 Nos)	3500 kW
11	BFP	594 t/hr or 165 kg/s
12	Economizer inlet temperature of water	243℃
13	Mass of steam leaving final super heater	595 t/hr or 153.7 kg/s
14	Mass of steam leaving Re heater	553.3 t/hr or 146.1 kg/s
15	CO_2 produced	3294.5 tons per day.

<u>Coal Analysis</u>:

Fixed carbon = 26 %

Volatile Matter = 22 %

Moisture = 10 %,

Ash = 42 %

Exergy Grade Factor (EGF) = 1.06 [32]

Furnace outer surface area = 1800 m^2

Mass of Air required for combustion = 5 kg/kg and excess air used is 30 %.

TABLE A.2. OPERATING PARAMETES OF TURBINE

Sub component	**Mass in kg/s**	**Temperature ℃**	**Pressure in bar**	**Specific Enthalpy in kJ/kg**	**Specific Entropy in kJ/kgK**
HP Turbine inlet	165.3	540	147.1	3425.3	6.40
HP Turbine outlet	153.7	350	40.47	3089.8	6.58
HP extraction (HP heater 6)	11.6	350	40.47	3089.8	6.58
IP Turbine Inlet	153.7	540	36.42	3539.8	7.71
IP Turbine outlet	122.85	315	5.88	30.91.2	7.74
IP Extraction (HP heater 5)	1.0	423	16.64	3302.2	7.21
IP Extraction (HP heater 4)	29.85	315	5.88	3091.2	7.74
LP Turbine inlet	122.85	315	5.88	3091.1	7.74
LP Turbine outlet	107.15	46	0.09	2400.0	7.62
LP Extraction (LP heater 3)	7.0	189	2.28	2841.7	7.30
LP Extraction (LP heater 2)	4.8	99	0.86	2659.4	7.35
LP Extraction (LP heater 1)	3.9	80	0.417	2618.5	7.50

TABLE A.3. OPERATING PARAMETES OF CONDENSER

Parameter	**Type/Numerical value**
Type	Surface type , single pass
Steam at condenser entry	46 ° C & 0.09 bar
Inlet and out cooling water Temp	26.62 ° C & 37.26 ° C
Tubes	19,208 ,Copper material
Copper Tube OD & ID	25.4 & 24.0 mm
Copper Tube Length	11.28 m
Condensate Heat load	221171743.8 Kcal/hr , 260 MW
Water Flow	21033.95 t/hr , 5842.76 kg/s
Load on Each condenser tube and heat transfer rate per unit area	13.5 kW and 15 kW/m^2

Appendix II

Calculation of Exergetic Efficiency for a Case Study

Understanding the Exergy

The word Exergy has been introduced by Professor Z.Rant (1956). The word Exergy made of two parts: ex- Latin prefix for "from or out" , akin to Greek ε (ex) or εξΟ (exo) and εργον (ergon) –greek for work [43,44,45] . This implication is that exergy is the work which can be obtained from a given system. Exergy is also known under the names - such as, available energy, availability, assergy and technical ability to do work. [29,30,31].

Exergy is a measure, that indicates to what degree energy is convertible to other forms of energy. (Refer Fig A.1)

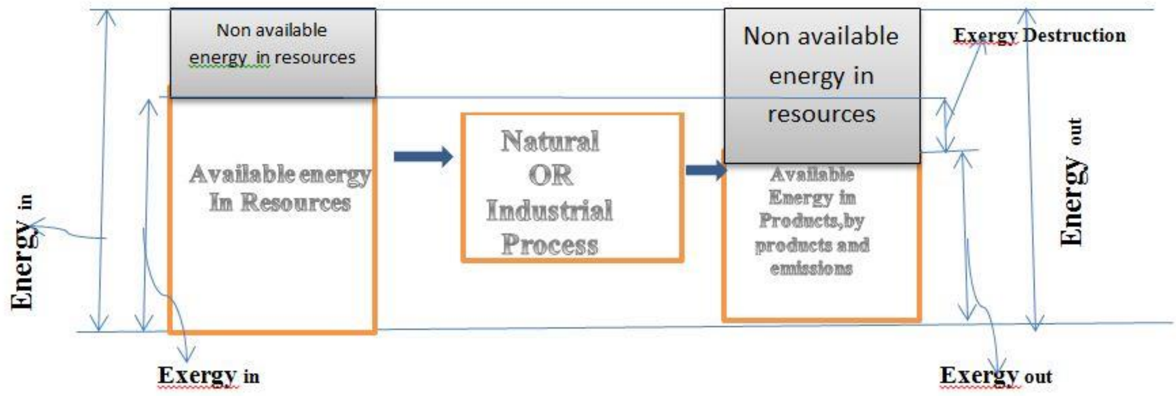


Fig A.1 Exergy Concept

For an ecosystem the total exergy is a measure of the change in entropy content from the equilibrium and the actual state. Thus,

Energy = Exergy + Anergy

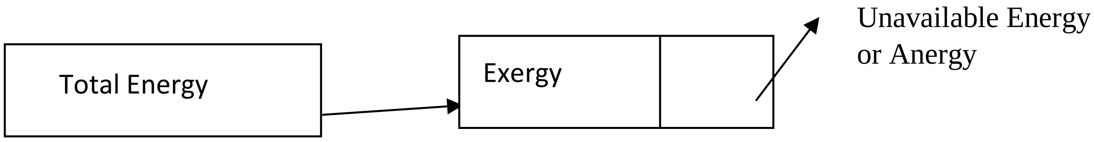
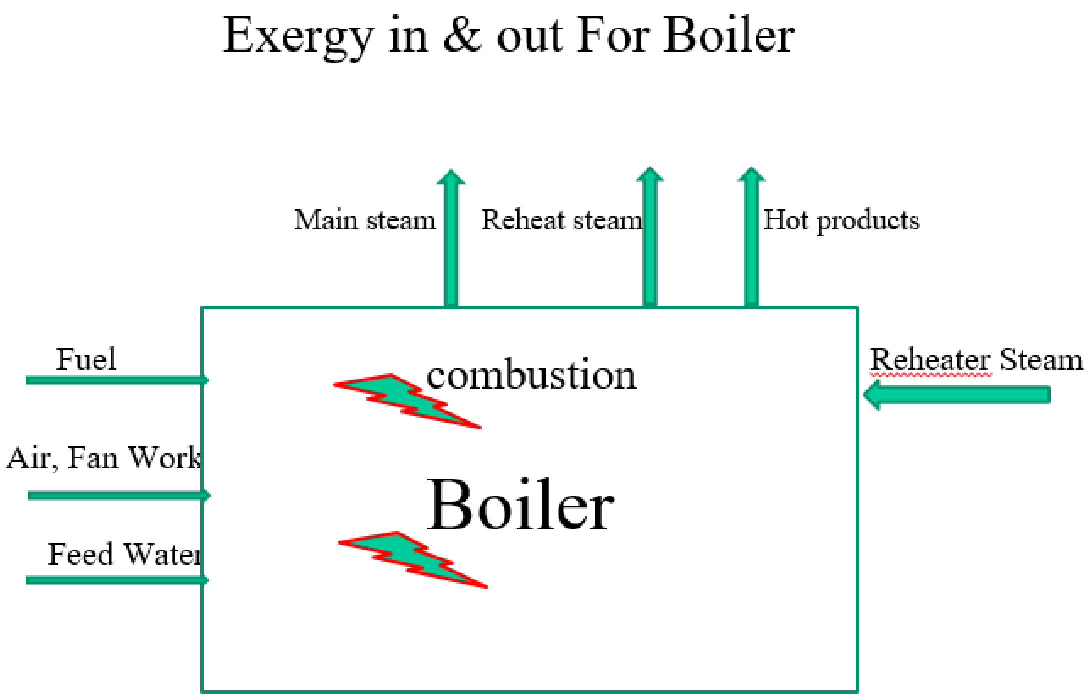


Figure A.2 . Representation of Energy, Anergy and Exergy in heat transfer.

1. **Boiler Exergy Calculations**



A. **Exegy inlet into the Boiler $Ǝ_{B,in}$**

i. Exergy supplied by the Fuel coal , $Ǝ_{Fuel}$

= Coal GCV x Coal Firing Rate x Exergy grade factor of fuel

= 14654.5 x 40 x 1.06 (kJ/kg)(kg/s)

= 621.3 MW

ii. Exergy supplied by Feed water, $Ǝ_{FW}$

$= \dot{m}_{FW} (CP)_{FW}. [(T_{FW} - T_0) - T_0 \ln (T_{FW}/T_0)]$

= 165.3 x 4.84 x [(243-25)-298ln (516/298)] (kJ/kg)(kg/s)

= 800 [218-298ln (1.73)] kJ/s

= 43.7 MW

iii. Exergy Due to inlet Air, $Ǝ_A$

Air from atmosphere enters APH through FD fan. APH is considered as part of boilers. Hence it is assumed the inlet air doesn't carry any exergy into the boiler. The outlet of APH is divided into Primary Air (PA) and Secondary Air (SA).

The exergy inlet is due to FD and PA fan work

The work input due to 2 FD fans and 2 PA fans.

$$= (2 \times 750 + 2 \times 1250) \text{ kW}$$
$$= (1500 + 2500) \text{ kW}$$
$$= 4 \text{ MW}$$

iv. Exergy due to Re-heater inlet (CRH) , $Ǝ_{CRH}$

$$= \dot{m}_{CRH}[(h_{CRH}-h_0) – T_0(S_{CRH} – S_0)]$$
$$= 153.7[(3089.8-104.9)-298(6.58-0.367)]$$
$$= 153.7[2984.9-1851.5]$$
$$= 174.2 \text{ MW}$$

Exergy input to boiler, $Ǝ_{B,in}$

$$Ǝ_{B,in} = Ǝ_{Fuel} + Ǝ_{FW} + Ǝ_{A} + Ǝ_{CRH} + \text{BFP work}$$
$$= 621.3+43.7+4+174.2+7$$
$$= 850.2 \text{ MW}$$

B. **<u>Exegy Exit of the Boiler $Ǝ_{BE}$</u>**

i. Exergy carried out by Main steam, $Ǝ_{MS}$

$$Ǝ_{MS} = \dot{m}_{MS}[(h_{MS} – h_o) – T_o(S_{MS} – S_o)]$$

$$= 165.3[(3425.3-104.9)-298(6.40-0.367)]$$
$$= 165.3[3320.4-298 \times 6.033]$$
$$= 251.7 \text{ MW}$$

ii. Exergy carried out by HRH, $Ǝ_{HRH}$

$$Ǝ_{HRH} = \dot{m}_{HRH} [(h_{HRH} – h_o) – T_o(S_{HRH} – S_o)]$$
$$= 153.7[(3539.8-104.9)-298(7.71-0.367)]$$
$$= 153.7(3434.9-2188.2)$$
$$= 191.6 \text{MW}$$

iii. Exergy carried out by Hot Products , Ǝ_{HPr}

$(c_p)_g$ = Specific heat of hot flue gas = 1.32

Temperature of flue gas = 136^0 C

Mass of flue gas = 300 kg/s

$$\text{Ǝ}_{HP} = \dot{m}_{HP}[(h_{HP} - h_o) - T_o(S_{HP} - S_o)]$$

$$= 300\text{x } 1.32\text{x}[(136\text{-}25)\text{-}298\ln(409/298)]$$

$$= 300\text{x}1.32[111\text{-}298\ln 1.37]$$

$$= 6.8 \text{ MW}$$

Now , Exery exit of the boiler,

i. $\text{Ǝ}_{B,out} = \text{Ǝ}_{MS} + \text{Ǝ}_{HRH} + \text{Ǝ}_{HP}$

$$= 251.7 + 191.6 + 6.8$$

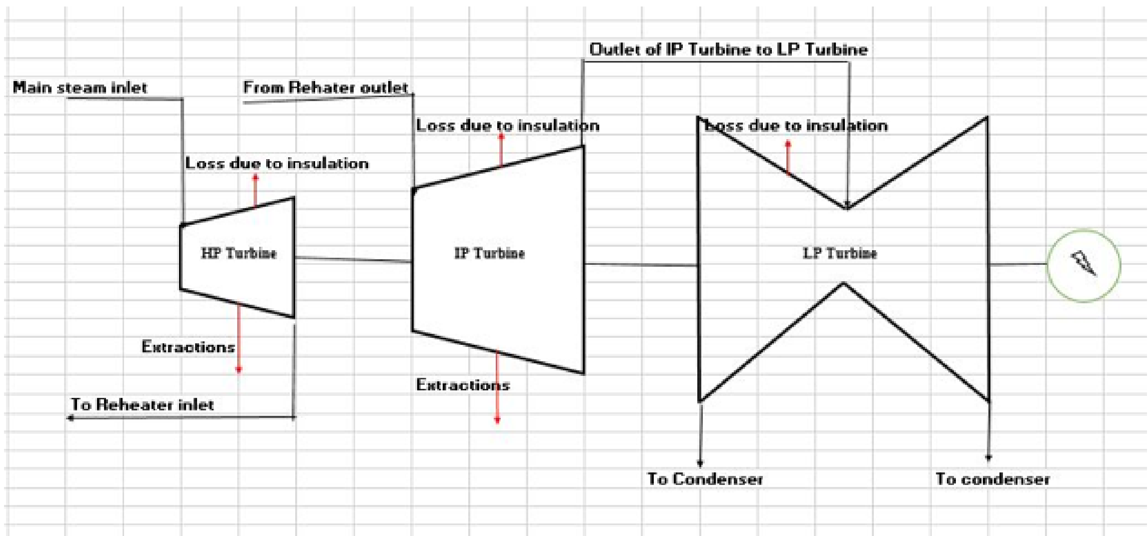
$$= 450.1 \text{ MW}$$

Exergy destruction in the boiler = $\text{Ǝ}_{B,D}$ = 850.2-450.1 = 400.1 MW

Exgergy efficiency of boiler = $\text{Ǝ}_{\eta B}$ = 450.1/850.2 = 52.9 % ≈ 53%

2. Exergy Analysis of Turbine

Exergy in and out for Turbine



i. <u>High Pressure Turbine</u>

Exergy carried into the HP turbine, $\exists_{HP,in} = \dot{m}_{HP,in}[(h_{hp.in} - h_o) - T_o(S_{HP,in} - S_o)]$

$= 165.3[(3425.3\text{-}104.9)\text{-}298(6.40\text{-}0.367)]$
$= 165.3[3320.4\text{-}298\text{x}6.033]$
$= 251.7$ MW

Exergy carried out by CRH, $\exists_{CRH} = \dot{m}_{CRH}[(h_{CRH} - h_o) - T_o(S_{CRH} - S_o)]$

$= 153.7[(3089.8\text{-}104.9)\text{-}298(6.58\text{-}0.367)]$
$= 174.2$ MW

Exergy carried out by Extraction (HP6),
$\exists_{HP6} = \dot{m}_{HP6}[(h_{HP6} - h_o) - T_o(S_{HP6} - S_o)]$

$= 11.6[(2984.9\text{-}104.9)\text{-}298(6.58\text{-}0.367)]$
$= 13.15$ MW

Work done by HP Turbine $= m_{HPH} \text{ x } h_{HPH} - m_{CRH} \text{ x } h_{CRH} - m_{HP6} \text{ x } h_{HP6}$

$= (165.3 \text{ x } 3425.3) - (153.7 \text{ x } 3089.8) - (11.6 \text{ X } 3089.8)$

$= 55.5$ MW

Exergy destruction in the HP Turbine $= \text{Ǝ}_{HPT,D}$

$= 251.7 – (174.2 + 13.15) – 55.5$

$= 8.85$ MW

ii. <u>Inter mediate Pressure Turbine</u>

Exergy carried into the IP turbine, $\text{Ǝ}_{IP,in} = \dot{m}_{ip,in} [(h_{ip.in} – h_o) – T_o(S_{ip,in} – S_o)]$

$= 153.7[(3539.8-104.9)-298(7.71-0.367)]$

$= 153.7[3434.9-2188.2]$

$= 191.6$ MW

Exergy carried out $\text{Ǝ}_{ip.o} = \dot{m}_{ip.o} [(h_{ip.out} – h_o) – T_o(S_{ip.out} – S_o)]$

$= 122.85[(3091.2-104.9)-298(7.74-0.367)]$

$= 122.85(2986.3-2197.1)$

$= 97$ MW

Exergy carried out by Extraction (HP5),

$\text{Ǝ}_{HP5} = \dot{m}_{HP5} [(h_{HP5} – h_o) – T_o(S_{HP5} – S_o)]$

$= 1[(3302.2-104.9)-298(7.21-0.367)]$

$= 1.2$ MW

Exergy carried out by Extraction (HP4),

$\text{Ǝ}_{HP4} = \dot{m}_{HP4} [(h_{HP4} – h_o) – T_o(S_{HP4} – S_o)]$

$= 29.85[(3091.2-104.9)-298(7.74-0.367)]$

$= 29.85 (2986.3-2197)$

$= 23.5$ MW

Work done by HP Turbine $= m_{ip.in} \times h_{ip.in} – m_{ip.out} \times h_{ip.out} – m_{HP5} \times h_{HP5} - m_{HP4} \times h_{HP4}$

$= (153.7 \times 3539.8) – (122.85 \times 3091.2) – (1 \times 3302.2)- (29.85 \times 3057.7)$

$= 69.6$ MW

Exergy destruction in the IP Turbine $= \text{Ǝ}_{IPT,D}$

$= 191.6 – 97-1.2-23.5-69.6$

$= 0.3$ MW

iii. <u>Low Pressure Turbine</u>

Exergy carried into the LP turbine, $\text{Ǝ}_{LP,in} = \dot{m}_{Lp,in} [(h_{Lp.in} – h_o) – T_o(S_{Lp,in} – S_o)]$

$= 122.85[(3091.1-104.9)-298(7.4-0.367)]$.
$= 122.85(2986.2-2095.834)$
$= 109.4$ MW

Exergy carried out $\text{Ǝ}_{Lp.o} = \dot{m}_{Lp.o} [(h_{Lp.out} - h_o) - T_o(S_{Lp.out} - S_o)]$

$= 107.15[(2400.2\text{-}104.9)\text{-}298(7.62\text{-}0.367)]$
$= 107.15(2295.1\text{-}2161.4)$
$= 14.3$ MW

Exergy carried out by Extraction (LP3),
$\text{Ǝ}_{LP3} = \dot{m}_{LP3} [(h_{LP3} - h_o) - T_o(S_{LP3} - S_o)]$

$= 7[(2841.7\text{-}104.9)\text{-}298(7.3\text{-}0.367)]$
$= 7(2736.8\text{-}2066) = 4.7$ MW

Exergy carried out by Extraction (LP2),
$\text{Ǝ}_{LP2} = \dot{m}_{LP2} [(h_{LP2} - h_o) - T_o(S_{LP2} - S_o)]$

$= 4.8[(2659.4\text{-}104.9)\text{-}298(7.35\text{-}0.367)]$
$= 4.8\ (2554.5\text{-}2080.1))$
$= 2.3$ MW

Exergy carried out by Extraction (LP1),
$\text{Ǝ}_{LP1} = \dot{m}_{LP1} [(h_{LP1} - h_o) - T_o(S_{LP1} - S_o)]$

$= 3.9[(2618.5\text{-}104.9)\text{-}298(7.5\text{-}0.367)]$
$= 3.9\ (2513.6\text{-}2125.6)$
$= 1.5$ MW

Work done by LP Turbine =

$= m_{Lp.in} \times h_{Lp.in} - m_{Lp.out} \times h_{Lp.out} - m_{LP3} \times h_{LP3} - m_{LP2} \times h_{P2} - m_{LP1} \times h_{P1}$

$= (122.85 \times 3091.1) - (107.15 \times 2400) - (7.0 \times 2841.7) - (4.8 \times 2659.4) - (3.9 \times 2618.5)$

$= 379.7\text{-}257.16\text{-}19.9\text{-}12.8\text{-}10.2$

$= 79.64$ MW

Exergy destruction in the HP Turbine $= \mathbf{Ǝ_{IPT,D}}$

$= 109.4 - (14.3+4.7+2.3+1.5+79.64)$

$= 6.96$ MW

Total work done by Turbine $= W_{HP} + W_{IP} + W_{LP}$

$= 55.5+69.6+79.64$

$= 204.74$

Plant output = 204.7x Turbine Efficiency x Generator efficiency

= 204.7x 0.95x 0.98

= 190.6 ≈ 191 MW

Exergy destruction in the Turbine $= \text{Ǝ}_{HPT,D} + \text{Ǝ}_{IPT,D} + \text{Ǝ}_{IPT,D}$

$$= 8.85 + 0.3 + 6.96 = 16.11 \text{ MW}$$

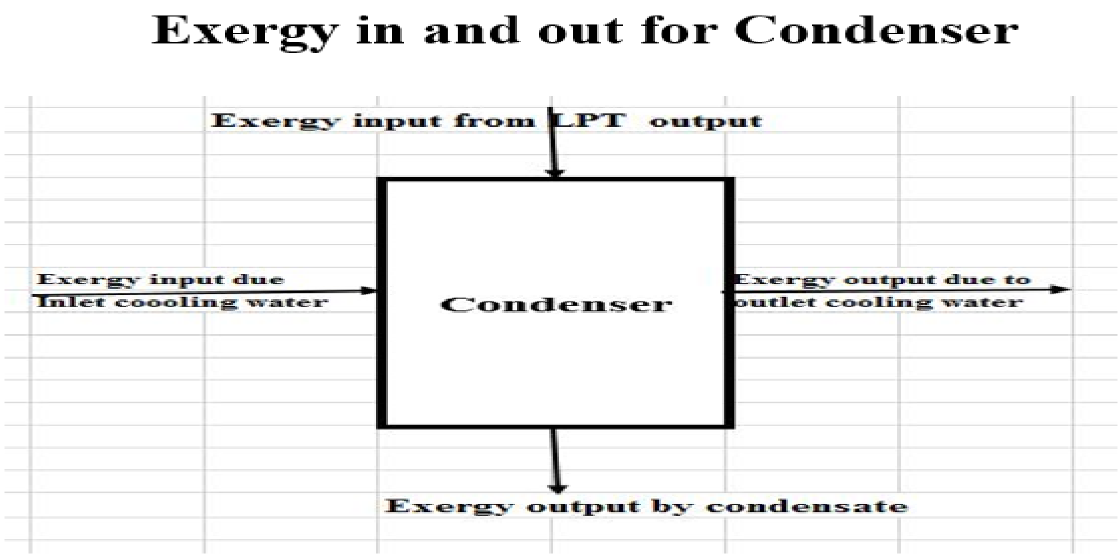
Exegetic Efficiency of the Turbine =

$$= \frac{Total\ Exergy\ inlet - Exergy\ Destruction}{Total\ Exergy\ inlet}$$

$$= \frac{(251.7+191.6+109.4)-16.11}{552.17}$$

$$= \frac{552.17-16.11}{552.17} = 97\%$$

3. **Exergy Analysis of Condenser**



Exergy inlet due to exhaust steam from LP Turbine = $Ǝ_{cond, s, in}$

$= \dot{m}_{steam,in} [(h_{steam,in} – h_o) – T_o(S_{steam,in} – S_o)]$

= 107.15[(2400-104.9)-298(7.6-0.367)]

= 107.15(2295.1-2161.4)

= 14.3 MW

Exergy entry due to cold water is $Ǝ_{cond, w, in}$

= 5842.76[111.8-104.84)-298(0.390-0.367)]

= 5842.76(6.96-6.854)

= 0.62 MW

Total Exergy inlet into condenser = $Ǝ_{cond,in}$ = 14.3 + 0.62 = 14.92 MW

Exergy destruction in the condenser = $Ǝ_{cond,d}$

$= T_{env} [CW \ln (T_{cl,out} /T_{cl,in}) + CW (T_{cl,in} – T_{cl,out})/T_{s,in}]$ [38]

Now , CW = Cooling water quantity = 5843 kg/s

T_{env} = Temp. Of the Environment = 25 ºC = 298 K

$T_{cl,out}$ = Cooling water out let Temp = 37.62 ºC = 310.62 K

$T_{cl,in}$ = Cooling water inlet Temp = 26.62 ºC = 299.62 K

$T_{s,in}$ = Steam inlet temperature = 46 ºC =319 K

Applying numerical,

Ǝ = 298 x 5843 x4.18 x[ln (310.62/299.62 + (299.62-310.62/319)]

= 298 x 5843 x 4.18 x (0.0361-0.0345)

= 11.65

Exergetic Efficiency of the Condenser = $\frac{14.92-11.65}{14.3}$ = 0.219 ≈ 0.22 %

Appendix III.

5E analysis Formulations and calculations for the case study

1 *Energy analysis:*

First law of Thermodynamics indicates that, the energy efficiency is ratio of work out to energy input.

$$\eta = \text{Work Output / Energy input.}$$

2 *Exergy analysis*

The exergy analysis is based on second law of thermodynamics and combines both first and second law of thermodynamics to evaluate both quality and quantity of energies (Kota 1980).In this analysis the expressions used for exergy calculations are mentioned below.

The exergy of a system ,Ǝ, at a specified state is given by the expression

$$\text{Ǝ} = (U-U_0)+ p_0(V-V_0) - T_0(S-S_0) + KE+ PE$$

Exergy balance for a closed system can be evaluated by,

$$\Delta\text{Ǝ} = change\ in\ exergy = \int_1^2 \left(1-\frac{T_0}{T_b}\right)\delta Q - [W - p_0\,(V_2 - V_1)] - T_0\Delta S$$

The exergy calculations for boiler,turbine and condenser of a thermal power plant are evaluated based on the above formulae.

3 ***Exergoeconomic Analysis:***

Exergoeconomics is a discipline which combines concepts of the Exergy method with those belonging to the economic analysis. The purpose of this exergoeconomic optimization is to achieve, within a given system structure, a balance between expenditure on capital costs and exergy costs which will give a minimum cost of the plan product.

According to Kotas and Szargut (1985) [32], the general expression for the capital cost of the plant is given by;

$$C = C_o + \text{Ǝ}_o\, k\{ \text{Ǝ}\eta / 1 - \text{Ǝ}\eta \}^m$$

Where C = capital cost of the plant

C_o= component of the capital cost which does not affect plant efficiency

Ǝ_o = Nominal plant exergy out put

Ǝ_η =rational efficiency of the plant = Exergy output/exergy input

k,*m* = Empiral constants which characterize a particular plant.

Exergoeconomic factor = $\Upsilon = \text{Ǝ}_o\,(\text{Ǝ}_{\eta\,1}/1-\text{Ǝ}_{\eta\,1})$

4. *Exergoenvironmental Analysis*

An exergo environmental analysis is an exergy-based method that identifies and calculates the location, magnitude, causes and environmental impact of thermodynamic inefficiencies in an energy conversion system. (Tsatsaronis and Morosuk, 2008). An exergo environmental analysis is also conducted at the component level of a system identifies to understand the relative importance of component with respect to environmental impact and options for reducing the environmental impact associated with the overall system. Oliverira (2013) reported that the Renewability exergy Index [36] takes into consideration the exergy associated to the useful products of a given energy conversion system. Mathematically, the relation can be expressed as ,

$$\text{Exergy Renewability index , } \exists_{RI} = \frac{\Sigma\, \exists_{Products}}{\exists_{Fuel} + \exists_{destroyed} + \exists_{deactivation} + \exists_{disposal} + \Sigma\exists_{emission}}$$

Depending on the value of the Renewability exergy index, it indicates that:

- Process with $0 \leq \exists_{RI} < 1$ are environmentally unfavourable.
- For internal and external reversible processes with non-renewable inputs, $\exists_{RI} = 1$
- If $\exists_{RI} > 1$, the process is environmentally favourable, and additionally, increasing $\exists_{RI}$ implies that the process is more environmentally friendly.
- When $\exists_{RI} \rightarrow \infty$, it means that the process is reversible with renewable inputs and no wastes are generated.

For thermal power stations, $\exists_{RI}$ is obtained as a function of the exergy efficiency of the power plant, taking into accounts that $\exists_{deactivation}$ and $\exists_{disposal}$ are zero . So the Exergy Renewability index equation can be re written as ,

$$\text{Exergy Renewability index , } \exists_{RI} = \frac{\text{Net output}}{\exists_{Fuel} + \exists_{destruction} + \exists_{Fluegas}}$$

5. Endurability Analysis

The environment enduring the pollution emitted by different industries and processes. One of major culprit is boiler emittance like solid particulates and heat energy. For the better environmental sustainability, it is essential to utilize non-renewable sources like coal, oil effectively in addition to use renewable sources. This will enable the mankind to maximize its use of limited resources and make existing resources last longer. Hence the Thermal power plants with low endurability factor will be continued in the use.
The endurability factor will be calculated as,

$$\text{Endurability factor} = Ə = \text{Exergy Destruction / Exergy input}$$

Hence, the environmental impact from the power plant can reduced by reducing the irreversible exergy losses of the plant.
By using above formulae the 5E parameters for the said case are calculated as below.

I. Efficiency (η)

η = Energy output / energy input

= (Energy out – Auxiliary consumption) / Calorific value of coal x coal consumption

= [(191-8.742) x 1000] / 14654.5 x 40

= 31.09 %

II. Exergy efficiency ($Ǝ_\eta$)

$Ǝ_{d,b}$ = Exergy desturction in the boiler = 400.1 MW

$Ǝ_{d,T}$ = Exergy desturction in the turbine = 16.11 MW

$Ǝ_{d,c}$ = Exergy desturction in the condenser = 11.65 MW

$Ǝ_{\eta,u}$ = 1- (Exergy Destruction/Exergy input)

= 1- (400+16.11+11.65)/850.2 = 49.7

III. Exergoeconomic Factor (ϒ)

$ϒ = Ǝ_{out} (Ǝ_{\eta\ 1}/1-Ǝ_{\eta\ 1})$ = 422.24 { 0.497/(1-0.497) } = 417.2

IV. **Exergy Renewability index ($Ǝ_{RI}$)**

$$Ǝ_{RI} = 191/(621.3+427.8+6.8) = 18\ \%$$

V. **Endurability factor (Ə)**

$$Ə = \text{Exergy destruction} / \text{Exergy input}$$

$$= (400.1+16.11+11.65)/621.3 = 0.689$$

Appendix IV

Design Calculations of Proposed Heat pipe for Condensation

The details of Heat Pipe designed for this purpose are as follows. (Fig A.3)

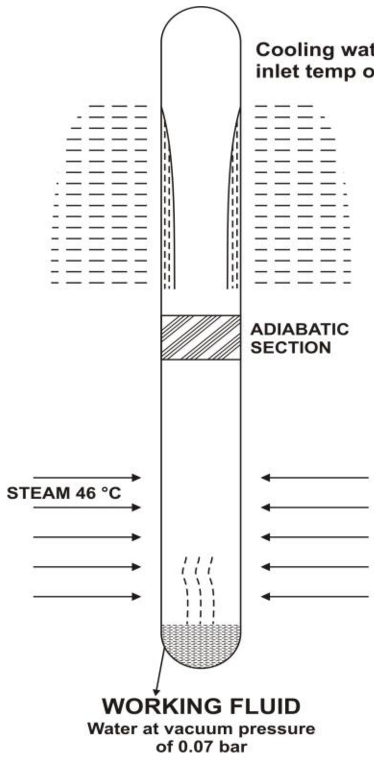


Fig A.3: Proposed Heat Pipe

The specifications of the heat pipe presented in Table 4.2

Thermodynamic Parameters of the proposed Heat Pipe can be calculated based on ref.[39,40,41]

Distilled water chosen as working fluid and compatible material for this is copper.

Aspect Ratio (AR) = L_e / d_i = 2/0.0497 = 40.24 ≈ 40

3.2 Calculation for the working Fluid inventory.

It is proposed to design the heat pipe such that, thermal Load on each heat pipe will be 30 kW

According to references, the quantity of working fluid, V_t, was calculated as below.

$V_t = [\ 0.8x(L_c + L_e) + L_a\ ]\ [\ 3Q_1\mu_l\,(\pi\, d_i)^2 / \rho_l^2 g h_{fg}\ ]^{1/3} = 1.4x\ 10^{-4}\ m^3$

The safety factor proposed was 7.865. accordingly, the working fluid will be,

$$= 1.4 \times 10^{-4} \times 7.865 = 1.1011 \times 10^{-3}\ m^3$$

The Filling Ratio (FR) = Volume of the Working Fluid / Volume of the Evaporator section

$$= 4 \times 1.1011 \times 10^{-3} / \pi (0.0497)^2 \times 2 \quad = 0.28 \text{ or } 28\ \%$$

3.3 BOILING LIMIT. The highest heat transfer quantity, Q_2, from boiling point of view, will be {According to Gorbis, Z. R & Savchenkov , G.A (1976)} [52]

$$Q_2 = Ku \{ h_{fg}\, \rho_v^{0.5} [\sigma\, g\, (\rho_l - \rho_v)]^{0.25} \}$$

$$\text{Where , } Ku = 0.0093\, (AR)^{-1.1} [d_i/L_e]^{-0.88}\, (FR)^{-0.74} (1 + 0.03\, Bo)^2$$

$$Bo = \text{Bond Number} = 0.0497\, \{9.81 \times (992 - 0.05)/69.6 \times 10^{-3}\}^{1/2}$$

$$= 18.6$$

Hence, $Q2 = Ku \{ h_{fg}\, \rho_v^{0.5} [\sigma\, g\, (\rho_l - \rho_v)]^{0.25} \}$

$$= 0.026\{2408 \times 10^3 \times 0.05^{0.5} [69.6 \times 10^{-3} \times 9.81\, (992 - 0.05)]^{0.25}\}$$

$$= 71417.86\ W \approx 71\ kW$$

3.4 FLOODING LIMIT. The highest transfer Quantity, from the flooding point of view, Q_3 will be given as,

$$Q_3 = K\, h_{fg} A_{cross} [g\sigma(\rho_l - \rho_v)]^{0.25} \times [\, \rho_v^{-1/4} + \rho_l^{-1/4}]^{-1/2}$$

$$\text{Now } K = [\rho_l / \rho_v]^{0.14} \tanh^2 (Bo)^{1/4}$$

$$K = 4 \times 0.942 = 3.77$$

And $Q_3 = 3.77 \times 2408000 \times 1.94 \times 10^{-3} \times 5.10 \times 0.66 = 59.3\ kW$

Summarizing the above results are given Table A4.

TABLE A.4. THERMODYNAMIC PROPERTIES OF DESIGNED HEAT PIPE OF CONDENSATION.

Sl.No	**Parameters**	**Desired requirements of Heat Pipes in the proposed HPHE**	**Designed Heat pipes characteristics as per different calculations**
1	Maximum heat transfer limit from the Boiling point of view	30 kW	71 kW
2	Maximum heat transfer limit from the Flooding point of view	30kW	59.3 kW

Heat Pipe Heat Exchanger Calculations

Calculations for Heat pipe loaded condenser (Fig A.4)

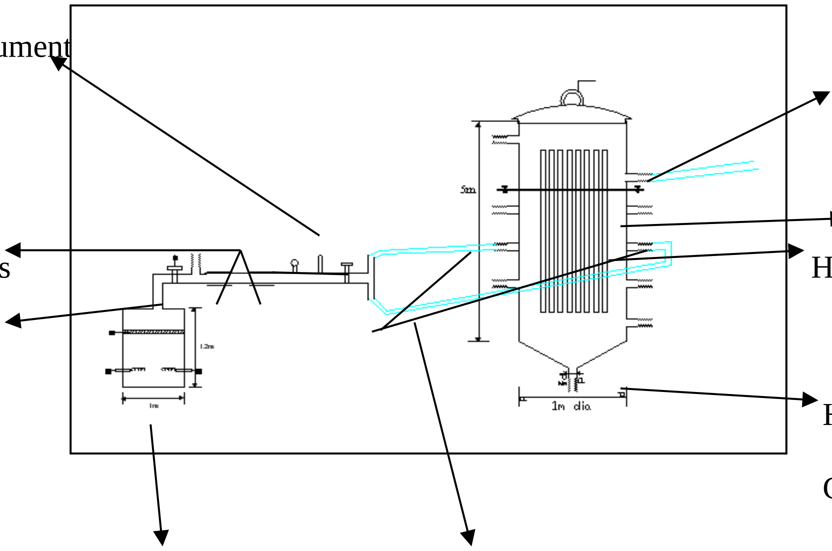


Fig A.4. Line Diagram of Experimental Set up

Steam inlet conditions into the condenser = 46 ° C, 0.09 bar (The values are from a running power plant)

Condenser load for 210 MW turbine, = 221171743.8 kW ≈ 260 MW

Calculation of Heat Transfer Coefficient (h_1) for the portion of heat pipe which is exposed to inlet steam

t_{sat}= steam inlet temperature =46 ° C

Steam inlet pressure = p=0.10 bar

μ_{l1} = viscosity of condensed water = 577×10^{-6} kg/m.s

h_{fg1}= heat of vaporization of inlet steam = 2390 kJ/kg

The saturated water temperature inside of heat at evaporator section = 39 ° C = t_s

Hence it can be safely assumed the wall temperature at evaporator section as 42° C

Hence the outside film temperature = t_f = (42 + 46)/2 = 44 ° C = 317 K

The modified heat of vaporization = $h_{fg1}{}' = h_{fg1} + 0.68\, C_{pl}(t_{sat} - t_s)$

$= 2390 + 0.68 \times 4.180 \times (46-40)$

$= 2407$ kJ/kg

According [47] the heat transfer coefficient correlation of two-phase region of the condenser,

$h_1 = h_{tp} = f(\chi_u) \{ k_f^3 \rho_f(\rho_f-\rho_g) g h_{fg1}/\mu_f . d_i .(t_{sat}-t_s)\}^{1/4}$

Where $f(\chi_u) = 0.375/\chi_u^{0.23}$

And ,χ_u= Lockhart-Martinelli Parameter $=\left[\frac{\rho g}{\rho f}\right]^{0.5} \left[\frac{\mu f}{\mu g}\right]^{0.1} \left[\frac{1-x}{x}\right]^{0.9}$

$k_f = 634 \times 10^{-3}$ w/m.k

$\mu_f = 631 \times 10^{-6}$ kg/m.s ,$\mu_g = 9.69 \times 10^{-6}$ kg/m.s

$\rho_f = 991.1$ kg/m^3 , $\rho_g = 0.056$ kg/m^3

x = Dry ness factor = 0.93

By applying numerical values, $\chi_u = 7.5 \times 10^{-3} \times 1.52 \times 0.097 = 1.11 \times 10^{-3}$

$f(\chi_u) = 0.375/ (1.1 \times 10^{-3})^{0.23} = 1.8$

Hence, $h_1 = 1.8 \times (9.43 \times 10^{16})^{0.25} = 31543$ W/m^2.k

Calculation of Heat Transfer Coefficient (h4) for the portion of heat pipe which is exposed to water Pool

The heat transfer coefficient for portion of heat pipe immersed in the cooling water pool can be calculated based on correlations suggested by Whitakar [53] and Zhukaushas A, [54] .

The Nusselt number for this , situation is,

$$Nu = C.\ Re_{2,max}^{m} .\ Pr^{0.36.} (Pr/Pr_s)^{1/4}$$

For $N \geq 20,\ 0.7 < Pr < 500,\ 1000 < Re_{max} < 2 \times 10^5$

Cooling water inlet temperature = $t_{c,in}$ = 26.62 ° C

Cooling water inlet velocity = V_{in}= 1 m/s

S_L = Longitudinal pitch = 0.07

S_T= Transverse pitch = 0.07

$$V_{max} = \{S_T/(S_T-d_o)\}V_{in} = 4.4 \text{ m/s}$$

Out let cooling water temperature = $t_{c,o}$ = 26.62+ (260 x 10^6/5843x 4179)
= 37.26 ° C

Hence average cooling water temperature = (26.62+37.26)/2 = 31.94 ° C= 304.94 K

D_{eq} for calculating Reynold number

$= 4 \{ S_L S_T – (\pi D_o^2/4)\}/ \pi D_o = 4 \{ 4.9 \times 10^{-3} - 2.3 \times 10^{-3}\}/0.169$

$= 0.0615$

Pr = 5.20, Prs = 4.62, $\mu_c = 769 \text{ X } 10^{-6}$ kg/m.s, $\rho_c = 995$ Kg/m^3 , k=620 x 10^{-3} w/mk

$Re_{2,max} = \{\rho_c V_{max} d_{eq}\}/ \mu_c = 3.5 \text{ X } 10^5$

Hence the constans for Euqation (5),

C = 0.022 , m = 0.84

Applying numerical to Euqation (5)

Nu = 1864.3

Hence , $h_4 = k \times Nu / d_o = 21405$ w/m^{2}K

Calculation of Heat Transfer Coefficient (h_2) for the evaporator section , inside of heat pipe

The heat transfer coefficient for evaporator section inside heat pipe can be calculated based on correlations suggested by Imura (1979) et. al .[55]

$$h_2 = 0.32 \left[(\rho_l^{0.65} k_l^{0.3} c_{pl}^{0.7} g^{0.2} q_e^{0.4})/(\rho_v^{0.25} h_{fg}^{0.4} \mu_l^{0.1})\right] x\, [p_{sat}/p_a]^{0.3}$$

ρ_l = liquid density inside heat pipe = 992 kg/m^3 . $c_{p.l}$ = 4178.5 J/kg.k

q_e= heat flux at evaporator section from outside = 30000/(π x 0.0540 x 2)

= 88419.4 w/m^2

ρ_v = 0.05 kg/m^3 ,h_{fg} = 2402 x 10^3 kg/m^3, μ_l = 663 x 10^{-6} kg/m.s ,k

= 631 x10^{-3} w/mk

therefore, applying numerical values , h_2 = 15,521 w/m^{2}k

Calculation of Heat Transfer Coefficient (h_3) for the condenser section , inside of heat pipe

The Reynold for condensate flow inside heat pipe is given by,

$$Re_3 = Q/\pi.d_o\mu_l h_{fg} = 30/ \pi.(0.047) \times 631\times 10^{-6} \times 2408 = 133.7$$

Hence the flow is laminar and the heat transfer coefficient for condenser section inside heat pipe can be calculated based on correlations suggested by Mc Adams correlation

,

$$h_3 = 1.13 \{ \rho_l g k_l^3(\rho_l - \rho_v)[h_{fg} + 0.68 \times c_{p.l}(T_{sat} - T_w)]/ (\mu_l d_i(t_{sat}-t_w)\}^{1/4}$$

t_w = surface wall temperature at condenser section that is part of heat pipe immersed in the water pool $\}$ = (26.62+39)/2 = 32.81 ° C = 305.8 K

the physical properties are evaluated taken at this wall temperature.

ρ_l = 995 kg/m^3 , ρ_v = 0.034 kg/m^3 , K_l= 620 x 10^{-3} w/m.k

h_{fg} = 2426 x 10^3 J/kg , $c_{p.l}$= 4178 j/kg.k , μ_l = 769 x 10^{-6}

Applying these numerical values ,

$$h_3 = 14065 \text{ w/m}^2\text{.k}$$

Calculation of Overall Thermal Resistance (U_0)
Heat Pipe inside Parameters

Inside area at evaporator section =$\mathbf{A_{ie}}$ = π d_iL = 0.312 m^2

Inside area at condenser section = $\mathbf{A_{ic}}$ = π d_iL = 0.312 m^2

Total inside area = A_i = 0.624 m^2

Heat Pipe outside Parameters

Outside area at evaporator section =$\mathbf{A_{oe}}$ = π d_oL = 0.34 m^2

Outside area at condenser section = $\mathbf{A_{oc}}$ = π d_oL = 0.34 m^2

Total outside area = A_o = 0.68 m^2

Calculation of overall internal heat transfer coefficient (U_i) of Heat Pipe

$$1/U_i.A_i= 1/h_3A_{ic} + 1/h_2A_{ie} = 2.3 \times 10^{-4} + 2.06 \times 10^{-4}$$

Applying numerical values, U_i = 3675.6 w/m^{2}K

Calculation of overall internal heat transfer coefficient (U_o) of Heat Pipe

$$1/U_o.A_o= 1/h_1A_{oe} + 1/h_4A_{oc} = 9.3 \times 10^{-5} +1.4 \times 10^{-4}$$

Applying numerical values, U_{out} = 6311.5 w/m^{2}K

Considering the outside surface, the overall heat transfer coefficient is (U) is ,

$$1/U A_o= 1/U_iA_i + 1/U_oA_o$$

Applying Numerical values, U = 2406 w/m^{2}K

LMTD calculation of Heat Pipe Condenser

Steam entry temperature = 46 ° C , and steam is condensing at this temperature.

Cooling water inlet temperature = 26.62 ° C ,

Cooling water outlet temperature = 36.65 ° C

LMTD = (19.38-9.35)/ ln {(46-26.62)/(46-36.65)} = 10.03/ln(19.38/9.35) = 13.5

Total Heat transfer area of single heat pipe =A = π (d_oL_e+ d_iL_e+d_oL_c+d_iL_c) = 1.3 m^2

Let N be number of heat pipes in the HPHE

Q = U_pNA (LMTD)

260 x 10^6 = 2406 x N x 1.3 x 13.5 implies , N = 6157.4 ≈ 6158

Effectiveness Comparison [42]

Conventional Condenser

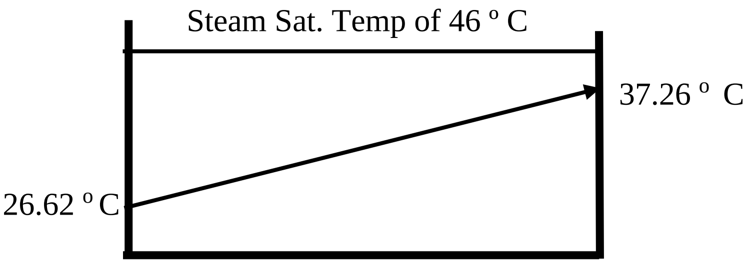


Fig. A.5 variation of temperatures in the conventional condenser

Condenser load = Q = 260,000 kW

Cooling Water Quantity = 5843 Kg/s

Conventional Condenser

Total Steam Load on the condenser= Q

= 260,000 kW

Cooling Water Quantity = 5843 Kg/s

Real Heat Transfer to the cooling water

= m c_p ($T_{cl,out}$– $T_{cl,in}$)

= 5843 x 4.178 x (37.26-26.62) = 259744.25 kW

Maximum possible heat transfer to the cooling water

= 5843 x 4.178 x (46-26.62) = 473105.60 kW

Effectiveness of the Existing Condenser =

Actual Heat Transfer/Max. Possible Heat Transfer

= 259744.25/473105.60

= 0.54 $\approx$ 55 %

Heat pipe loaded condenser

With ref. [40], the heat pipe condenser can be considered as liquid-coupled, indirect-transfer-type exchanger system. The analysis carried out as per ref. [42,43] .

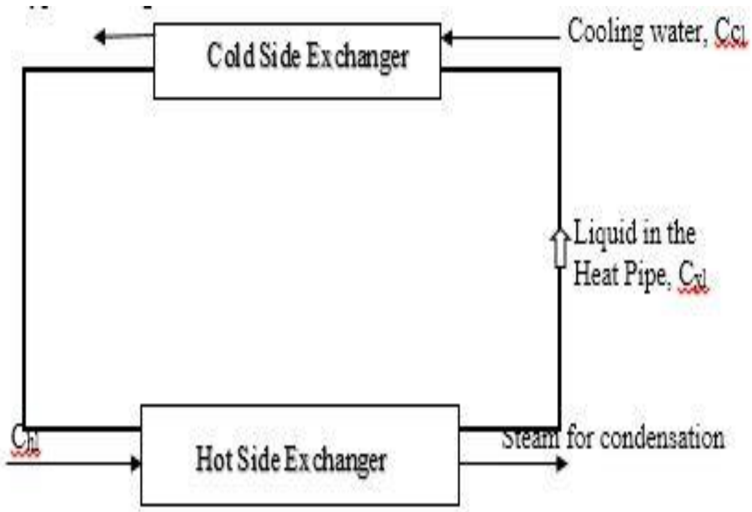


Fig A.6. Equivalence between HP condenser and liquid coupled indirect Heat Exchanger

C = mass flow x specific heat =Flow Stream capacity, kcal/s.K, Subscripts xl, cl, hl are coupling fluid, cold fluid, hot fluid respectively.

c_p is specific heat of the fluid in kcal/kg.K

C_{cl} = Mass of cooling water x $c_{p,c}$ = (5843/9025) x 4.187 = 2.71 kJ/s.K

(Assuming cooling water distributed equally to all heat pipes)

C_{xl} = m_{xl} x $c_{p,xl}$ = mass of water used inside the heat pipe for the purpose

= 0.0125 x 4.178 = 0.0522 kJ/s.K

C_{hl} =m_{hl} x $c_{p,hl}$ = mass of steam condensed on each heat pipe x specific heat of steam

= 0.012 x 1.895 = 0.023 kJ/s.K

Now , Evaporator section of Heat pipe, $NTU_e = U_e\pi\, DL_eN / m_e.c_p$, Now for **evaporator section**, NTUe $=\frac{UeAe}{mcp}$

Where h_e = total thermal conductance of evaporator section (W/K)

A_e = outer area of evaporator section(m^2)

m = mass flow rate of steam (Kg/s)

Thermal resistance of evaporator section = resistance due to steam entering+ resistance due to wall+ resistance inside heat pipe

$$Re = Ro,e + Rw,e + Ri,e$$

The Resistance R will be calculated as reciprocal of product of heat transfer coefficient and area. Accordingly,

$R_{o,e} = 9.32 \times 10^{-5}$ K/W

$R_{w,e} = 1.67 \times 10^{-5}$ K/W

$R_{i,e} = 2.0 \times 10^{-5}$ K/W

$R_e = 1.3 \times 10^{-4}$ K/W

Total thermal conductance of evaporator section = (1/Re) = 7698.2 W/K

NTUe = (7698.2 x 0.34/0.012 x 1.895 x 10^3) = 115.10

Effectiveness of evaporator section is $\varepsilon_e = 1- e^{-NTUe} = 1-e^{-115.10} = 1$

For condenser section

Thermal resistance of condenser section = resistance due to steam entering+ Resistance due to wall+ resistance inside heat pipe

$$R_c = R_{o,c} + R_{w,c} + R_{i,c}$$

$R_{o,c} = 1.59 \times 10^{-5}$ K/W

$R_{w,c} = 1.67 \times 10^{-5}$ K/W

$R_{i,c} = 2.21 \times 10^{-5}$ K/W

$R_c = 5.47 \times 10^{-5}$ K/W

Total thermal conductance of condenser section (W/K)

$$=(1/R_c) = 18{,}281.5 \text{ (W/K)}$$

NTU_c = (2570.7 x 0.34 / 0.65 x 4.187 x 10^3) = 2.3

Effectiveness of condenser section is $\varepsilon_c = 1- e^{-NTUc} = 1-e^{-2.3} = 0.89$

For a heat pipe heat exchanger with n rows of heat pipes [44]

For evaporator section, with n number of rows of heat pipes,

The net effectiveness , $\sum\varepsilon_e = 1- (1-\varepsilon_e)^n$

For condenser section, with n number of rows of heat pipes,

The net effectiveness $\sum\varepsilon_c = 1 - (1-\varepsilon_c)^n$

The overall effectiveness of the heat exchanger ε_o is given by,Ref [42]

$\varepsilon_o = \{ 1/\sum\varepsilon_e + (C_{hl}/C_{cl}) \times (1/\sum\varepsilon_c - 1) \}^{-1}$,since $C_{cl} > Cx_l > C_{hl}$

Appling numerical, $\varepsilon o \approx 0.99$

Exergy calculations of Conventional and Heat pipe based condenser

For Conventional Condenser

Exergy inlet due to exhaust steam from LP Turbine = $Ǝ_{cond, s, in}$
= 14.3 MW

Exergy entry due to cold water is $Ǝ_{cond, w, in}$ = 0.62 MW

Total Exergy inlet into condenser = $Ǝ_{cond,in}$ = 14.3 + 0.62 = 14.92 MW

Exergy destruction in the condenser = $Ǝ_{cond,d}$ = 11.65

Exergetic Efficiency of the Condenser = $\frac{14.92-11.65}{14.3}$ = 0.219 or 0.22 %

The Exergy Calculation of heat pipe based proposed condenser will be as follows

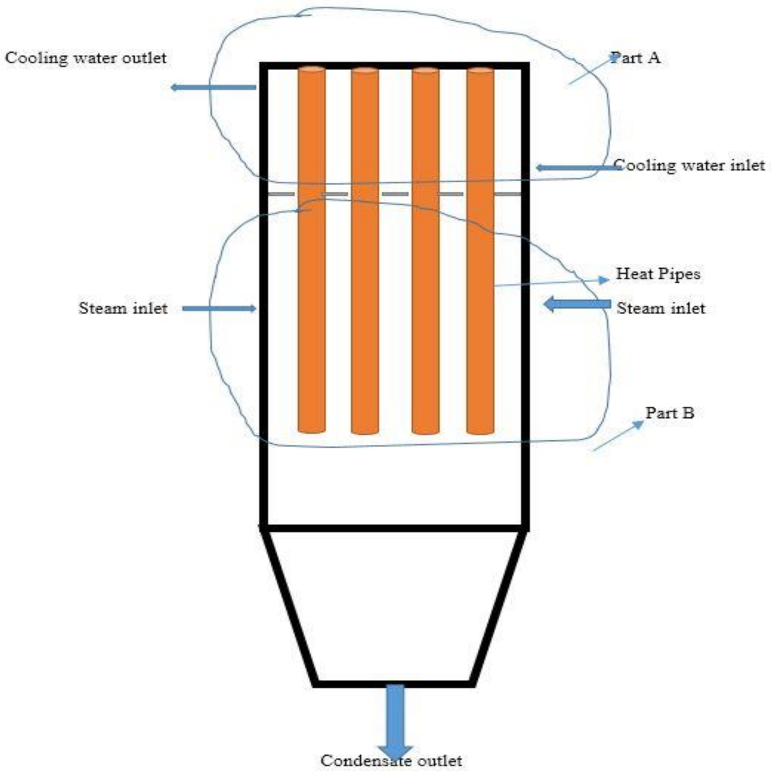


Figure A.7 . Line diagram of heat pipe condenser

<u>For Part B</u>

In this section steam condensates and liquid inside heat pipe gets evaporated.

$\Delta\exists=U_oA_o(\pi_T-1)^2/\pi_T]$

U_o=Overall heat transfer coefficient=2406 $W/m^2.K$

A_o=Overall heat transfer area=1.3 m^2

π_T=the ratio of input thermodynamic temperature of the streams=46/39.02=1.179

Applying numerical,

Hence, $\Delta\exists=2406x1.3x\ (1.179-1)^2/1.179=85$ W

Where steam condenses into water and fluid inside the heat pipe evaporates.

$\Delta\exists=U_oA_o(\pi_T-1)^2/\pi_T]$]

U_o=Overall heat transfer coefficient=2406 $W/m^2.K$

A_o=Overall heat transfer area=1.3m^2

π_T=the ratio of input thermodynamic temperature of the streams=46/39.02=1.179

Applying numerical,

Hence, $\Delta\exists=2406\times1.3\times(1.179-1)^2/1.179=85$ W

<u>For Part A</u>

In this section temperature of cooling water rises and vapour inside heat pipe will be condensate

.

$\Delta\exists=T_{env}[C_1\ln(T_1^{''}/T_1^{'})+C_1(T_1^{'}-T_1^{''})/T_2^{'}]$

C_c=Heat capacity of water stream, W/k

$T_1^{'}$=cooling water at inlet=299.62 K 302.62

$T_1^{''}$=cooling water at outlet=310.26 K 312.65

$T_2^{'}$=vapor temp. Inside heat pipe before condensation=312.02 K

Hence,

$\Delta\exists=301\times5843\times4.178\times[\ln(310.26/299.62)+(299.62-310.26)/312.02]=7348028.3\times(0.0349-0.0341)=5878.4$ kW

Total Exergy in Part A and Part B=5878.4 kW+85 kW=5963.4 ≈5.9 MW

Appendix VI

Calculation of Exergy destruction of Boiler and Turbine based on Insulation properties

According to Central Electricity Authority of India, the boilers and Turbine are insulated with 75 mm thickness. The composition, thermal conductivity and overall resistance are given in the Table A.5.

TABLE A.5. BOILER INSULATION DATA

	Thickness of inner fire Brick layer	Thickness of ceramic blanket	Thickness of Galvanised steel sheet	Total
Thickness in mm	50	24	1	75
Thermal conductivity in w/mK	0.4	0.12	55	0.325

Boiler (For the case study boiler)

Total boiler outside wall surface area = 1800 m^2

Average boiler inside temperature = 1000 ^oC

The skin of boiler is covered by insulation which is designed According to ASTMC 680 standards [37] so that surface temperature will be at 60℃.

The heat lost from the boiler outside surface walls $= \frac{\Delta T}{\Sigma R} = \frac{1000-60}{0.325/1800}$ = 5.2 MW

Hence the exergy transferred to environment = [1- 298/(273+60)]x 5.2

= 0.546 MW

As per proposal if insulation properties are increased, so that the if, boiler wall surface is maintained at 40℃, then exergy transferred will be as follows.

The heat lost from the boiler outside surface walls $= \frac{\Delta T}{\Sigma R} = \frac{1000-40}{0.325/1800}$ = 5.32 MW

Hence the exergy transferred to environment = [1- 298/(273+40)]x 5.32

= 0.255 MW

Improvement in Exergy destruction saving = $\frac{0546-0.255}{850.2}$ = 0.03 %

Turbine (For the case study boiler)

The exergy loss for the turbine are as per Table A.6

The formulae used for heat transfer to environment and exergy transferred to environment are same as to boiler calculations.

TABLE A.6 . TURBINE INSULATION IMPROVEMENT CALCULATIONS

	Surface area	**Average inside temperature**	**Heat Transferred to environment**	**Exergy loss to environment**	**Heat Transferred to environment**	**Exergy loss to environment**
			When surface temp maintained at 60 ^{o} C		When surface temp maintained at 40 ^{o} C	
High pressure turbine hood	5.31 m^2	445 ^{o}C	6.3×10^{-3} MW	6.615×10^{-4} MW	6.62×10^{-3} MW	3.2×10^{-4} MW
High pressure turbine hood	18.7 m^2	427.5 ^{o}C	0.02 MW	2.1×10^{-3} MW	0.22 MW	1.06×10^{-3} MW
High pressure turbine hood	50.4 m^2	180.5 ^{o}C	0.019 MW	2.0×10^{-3} MW	0.22 MW	1.06×10^{-3} MW
Total				4.76×10^{-3} MW		2.4×10^{-3} MW

Appendix VII

The details of Proposed Heat Pipe designed for Platen super heater & connecting header and Pressure drop calculations.

The calculation philosophy and procedure is based on Ref . [39, 40, 41, 45]

Length of the Heat pipe = L = 13.05 m

Length of evaporator section = L_e = 12 m

Length of the condenser section = L_c = 1 m (This section of heat pipe consists fins)

Adiabatic section length = l_a = 0.05 m

Tube material is 316L SS tubes

Heat pipe outside diameter = d_o = 90 mm =0.09 m

Heat pipe inside diameter = d_i=80 mm = 0.08 m

Thickness of the heat pipe wall = t = 5 mm = 5.0×10^{-3} m

The heat pipe is a wickless, gravity assisted that is strictly it is a two phase closed thermosyphon

Heat pipe vacuum = 0.15 bar

Working fluid is Sodium metal. The properties are as reported in Table A.7

TABLE A.7. PROPERITES OF SODIUM METAL

Property	Numerical value
Boiling point at 1 atm pr	882 ^{o} C
Boiling point at 0.15 bar	700 ^{o} C
Density	971 kg/m^3
Thermal diffusivity	113.60×10^{-6} m^2/s
Specific heat	1206 J/kgK
Thermal Conductivity	109.3 w/mk

Thermodynamic Limits of Suggested Heat Pipe

The working fluid is selected as sodium [40]

As per Amir Faghri [40], the compatible material for sodium is stainless steel. 316L SS tubes proposed which is suitable for boiler application.

Now, Aspect Ratio (AR) = L_e / d_i = 12/0.08 = 150

Quantity of Working Fluid

The Heat pipes are to be designed for load of 130 kW

the quantity can be , $V_t = [\ 0.8x(L_c + L_e) + L_a\ ]\ [\ 3Q_1\mu_l\,(\pi\ d_i)^2 / \rho_l^2 g h_{fg}\ ]^{1/3}$

Q_1= load on proposed heat pipe = 130000 W
μ_l =liquid viscosity inside the heat pipe =1.9 x 10^{-5} kg/m.s

[Sodium fluid properties are evaluated at 700 ^{o} C]

g = acceleration due to gravity = 9.81 m/s^2

h_{fg} =latent heat of evaporation of water inside heat pipe J/kg = 4090 x 10^3 J/kg

ρ_l = liquid density inside heat pipe = 763.5 kg/m^3

Hence ,

$V_t = [\ 0.8x(1 + 12) + 0.05\ ]\ [\ 3x\ 130000 \text{ x } 1.9 \text{ x } 10^{-5}(\pi \text{ x } 0.08)^2 / (763.5)^2 \text{ x } 9.81 \text{ x } 4090 \text{ x } 10^3\ ]^{1/3}$

$= 10.45x2.714 \text{ x } 10^{-5} = 2.84 \text{ x } 10^{-4} \text{ m}^3$

Keeping the safety factor margin, it is decided to have 7.865 times of this quantity.

So, volume of sodium to be used = 2.84 x 10^{-4} x 8.5 = 2.4x 10^{-3} m^3

Filling Ratio = 4x 2.4 x 10^{-3} / π $(0.08)^2$ x 12 = 0.04 or 4.0 %

<u>BOILING LIMIT</u>

$$Q_2 = Ku\ \{\ h_{fg}\ \rho_v^{0.5}[\sigma\ g\ (\rho_l - \rho_v)]^{0.25}\}$$

$$Ku = 0.0093\ (AR)^{-1.1}[d_i/L_e]^{-0.88}\ (FR)^{-0.74}(1 + 0.03\ Bo)^2$$

Bo = Bond Number = $d_i\{\ g\ (\rho_l - \rho_v)/\sigma\}^{1/2}$

σ = surface tension N/m =1.33 x 10^{-2} N/m ,ρ_l = 763.5 kg/m^3 , ρ_v = 0.05 kg/m^3

With values of heat pipe inside,

$Bo = 0.08\ \{\ 9.81 \text{ x } (763.5 - 0.05)/1.33x\ 10^{-2}\}^{1/2}$

$= 60$

$Ku = 0.0093\ (150)^{-1.1}[0.08/12]^{-0.88}\ (0.04)^{-0.74}(1 + 0.03 \text{ x } 60)^2$

$= 3.8 \text{ x } 10^{-5} \text{ x } 82.2 \text{ x } 10.83 \text{ x } 7.84$

$= 0.265$

Hence ,$Q_2 = Ku\ \{\ h_{fg}\ \rho_v^{0.5}[\sigma\ g\ (\rho_l - \rho_v)]^{0.25}\}$

$= 0.265\{\ 4090 \text{ x } 10^3 \text{ x } 0.05^{0.5}[1.33 \text{ x } 10^{-2} \text{ x } 9.81\ (763.5 - 0.05)]^{0.25}\}$

$= 0.265 \text{ x } 914551.8x\ 3.16$

$= 765845.7 \text{ W} = \approx 766 \text{ kW}$

FLOODING LIMIT

Q_3 = Quantity of heat with reference to flooding limit

$$= K\, h_{fg} A_{cross} [g\sigma(\rho_l - \rho_v)^{0.25}] \times [\rho_v^{-1/4} + \rho_l^{-1/4}]$$

$$\text{Now } K = [\rho_l / \rho_v]^{0.14} \tanh (Bo)^{1/4}$$

Applying Numerical,

$K = 3.8$, And hence $Q_3 = 155$ kW

The thermodynamic details can be summarized to the Table A.8

TABLE –A.8. THERMODYNAMIC DETAILS OF PROPOSED HEAT PIPES.

Sl.No	**Parameters**	**Desired requirements of Heat Pipes in the proposed HPHE**	**Designed Heat pipes characteristics as per different calculations**
1	Limit of heat transfer from the Boiling point of view	130 kW	766 kW from (3)
2	Limit of heat transfer limit from the Flooding point of view	130 kW	155 kW from (4)

The above heat pipes can be used for the platen super heater safely.

Total number of heat pipes proposed = 58 x 6 = 348

The finned tubes are arranged as 6 x 58 , that is 6 tubes in width and 58 rows in depth in the direction of steam flow. The height of fin tube is 1 m. The fins are of SS 316 L material same as tube material. The dimensions of the heat pipes are arrived based on the space available in the existing boiler.

The Fin Details

N_f = Number of fins = 200 fins/m = n

Thickness of fin = b = 3mm = 0.003 m

The fin height = h = 25 mm = 0.025 m

Outer Diameter of tube = $d = d_o$ = 0.090 m

Base diameter of fin = outer diameter of tube = d=0.090 m

$S_L = S_T$ = Tube pitch = 190 mm = 0.19 m

S = Clearance between fins = (1/n - b) = 1/200 – 0.003 = 2×10^{-3} m

Calculations of the Heat Transfer coefficients

Flue gas inlet temperature = 1236 ° C as per boiler collected data.

<u>Determination of Heat Transfer Coefficient (h_1) for the portion of heat pipe which is exposed to inlet Flue gas</u>

The outside gas heat transfer coefficient h_1 is the sum of convective heat transfer coefficient h_c and non luminous heat transfer coefficient h_N.

Hence , $h_1 = h_c + h_N$

Now calculation of h_c

Now , According [45] $h_c = 0.33\ Re^{0.66} \times Pr^{0.33}$

$Re = G d/\mu$ G = Gas mass velocity kg/m^2s $= W_g / N_w L(S_T - d)$

W_g = gas flow over tubes = 283.2 kg/s

d = heat pipe outside Diameter = 0.09 m

L = Length of the Tube = 12

N_w = Number of tubes/rows =6

S_T, S_L = Transverse and longitudinal pitch = 0.19m

μ = Viscosity of gas , kg/ms or Pas = 53×10^{-6}

By applying numerical, G = 283.2/6 x 12x (0.19-0.090) = 39.3

$Re = 39.3 \times 0.090 / 53 \times 10^{-6} = 66{,}735.85$

Now, $Pr = \mu c_p / k = (53 \times 10^{-6} \times 1.34 \times 10^3) / 0.081 = 0.877$

Now , $Nu = 0.33 \times (66735.85)^{0.6} \times (0.877)^{0.33} = 247.9$

But $Nu = h_c \times d / k$ or $h_c = Nu \times k/d = 247.9 \times 0.081/0.09 = 223.11\ w/m^2\ k$

Calculation of h_N (Non Luminous heat transfer coefficient)

From Ref . V. Ganaphty, $h_N = \sigma\varepsilon_g(T_g^4-T_o^4)/(T_g-T_o)$

T_g = average gas temperature = 1236° C

T_o = Wall temperature = 968 ° C

Steaffan Boltzman constant = σ= 5.67 x 10^{-8} w/m^2k

From Reference,[45] $\varepsilon_g = 0.128$

Appling numerical , $h_N = 5.67 \times 10^{-8} \times (1236^4-968^4) / (1236-968) = 39.4\ w/m^2k$

Hence, $h_1 = h_c+h_N = 223.11+39.4 = 262.5\ w/m^2.k$

Determination of Heat Transfer Coefficient (h_2) for the evaporator section , inside of heat pipe

$$h_2 = 0.32 \left[(\rho_l^{0.65} k_l^{0.3} c_{pl}^{0.7} g^{0.2} q_e^{0.4})/(\rho_v^{0.25} h_{fg}^{0.4} \mu_l^{0.1})\right] x\ [p_{sat}/p_a]^{0.3} \quad \text{--- (6)}$$

ρ_l = liquid density inside heat pipe = 763.5 kg/m^3 . $c_{p.l}$ = 1273.0 J/kg.k ,k_l = thermal conductivity of liquid = 60.81 w/m.k

q_e= Quantity of heat at evaporator section = 130000/(π x 0.080 x 12) = 43104.5 w/m^2

ρ_v = 0.05 kg/m^3 ,h_{fg} = 4090 x 10^3 kg/m^3, μ_l = 1.9 x 10^{-5} kg/m.s

The equation 6, with numerical, $h_2 = 853.13\ kw/m^2K$

Determination of Heat Transfer Coefficient (h_3) for the condenser section , inside of heat pipe

The Reynold for condensate flow inside heat pipe is given by,

$$Re = Q/\pi.d_i\mu_l\, h_{fg} = 130/\ \pi. (0.080) \times 1.9 \times 10^{-5} \times 4090 = 6656.20$$

This indicates that, condensate flow is turbulent. According [46], heat transfer coefficient is,

$$h_3 = (0.0076)\, Re^{0.4} \{\rho_l\, g k_l^{\ 3} (\rho_l - \rho_v)]/ \mu_l^2\}^{1/3}$$

Appling physical properties for the wall temperature,

ρ_l = 765.3 kg/m^3 , ρ_v = 0.05 kg/m^3 , K_l= 60.81 w/m.k

μ_l = 1.9 x 10^{-5}

Applying these numerical to the equation 7,

$$h_3 = 3724.7\ kw/m^2.k$$

Determination of Heat Transfer Coefficient (h4) for the portion of heat pipe which is exposed to Steam

Convective heat transfer coefficient

According [47] for solid fins and for inline bundle arrangement,

The convective heat transfer coefficient = h_c =

$C_1C_3C_5\{(d+2h)/d\}^{0.5}\{T_g/T_f\}^{0.5}GC_p\{K/\mu/C_p\}^{0.67}$

Where G = W_g / [$N_wL(S_T-A_0)$]

W_g = steam flow , kg/s = 165.3

Obstruction area = A_0 = d + 2nbh = 0.090+ 2x 200 x 0.003 x 0.025 = 0.12 m^2/m

N_w = Tubes/rows or number tubes wide

L = Effective length,m = 1m

μ = viscosity of steam kg/m.s or Pa.s =2.517 x 10^{-5} Pas

C_p = Steam specific heat at average temperature = 3458.6 J/kgK

And k=thermal conductivity of steam = 0.0816 w/mk

Now C_1= 0.053 $[1.45-(2.9S_L/d)^{-2.3}]Re^{-0.21}$

G = 165.3/[6x1x(0.19-0.12)]= 165.3/0.42 = 393.6 kg/m^2s

Re = Gd/μ = 393.6 x 0.090/2.517 x 10^{-5} = 1407389.75

C_1 = $0.053[1.45-(2.9x0.19/0.090)^{-2.3}][1407389.75]^{-0.21}$

= 0.053 x 1.43x 0.051

= 3.87 x 10^{-3}

$C_3 = 0.20 + 0.65e^{-0.25h/s}$ = $0.20+0.65\ e^{-0.25x0.025/0.002}$ =0.23

$C_5 = 1.1-(0.75-1.5e^{-0.7x\ N_d})e^{-2.0\ x\ S_L/S_T}$ = $1.1-(\ 0.75-1.5\ e^{-0.7x\ 58})\ e^{-2.0x1}$

= 1.1 – $(0.75)e^{-2.0}$ = 0.9985

$C_2 = 0.11+1.4Re^{-0.4}$ = $0.11+\ 1.4\ (1407389.75)^{-0.4}$ = 0.115

$C_4 = 0.080(0.15S_T/d)^{-1.1(h/s)0.15}$ =$0.080[0.15x0.19/0.090]^{-1.1(.025/.002)0.15}$

= $0.080x(0.316)^{-1.607}$ = 0.509

C_6= $1.6 – (0.75-1.5e^{-0.7N_d})\ e^{-2.0(S_L/S_T)^2}$ = $1.6 – (0.75-3.5x10^{-18})$e-2.0

$= 1.6\text{-}(0.75)\text{x } 0.135 = 1.498$

Assuming the average fin temperature as 600^0 C,

Convective heat transfer coefficient, $h_c = C_1C_3C_5\{(d+2h)/d\}^{0.5}\{T_g/T_f\}^{0.5}GC_p\{k/\mu/C_p\}^{0.67}$

$= 3.87 \text{ x } 10^{-3} \text{ x } 0.23\text{x } 0.9985\text{x } \{ (0.090+ 2\text{x}0.025)/0.090\}^{0.5} \text{ x } (713/873)^{0.5}\text{x } 393.6\text{x } 3458.6 \text{ x } [0.0816/2.67433\text{x } 10^{-5} /3458.6]^{0.67}$

$= 8.9\text{x } 10^{-4} \text{ x } 1.247\text{x}0.904\text{x}1361304.96\text{x}0.9195 = 1255.83 \text{ w/m}^2\text{k}$

Efficiency For Solid Fins, In Line Arrangement

A_f = Area of fins per unit length = π n x [2 dh+2h^2+bd+2bh]

$= \pi \text{ x } 200 \text{ x } [2\text{x}0.090\text{x}0.025+2(0.025)^2+0.003\text{x}0.090+2\text{x } 0.003\text{x}0.025]$

$= \pi \text{ x } 200\text{x } [4.5 \text{ x } 10^{-3} + 1.25 \text{ x } 10^{-3} + 2.7 \text{ x } 10^{-4} + 1.5\text{x}10^{-4}]$

$= \pi \text{ x } 200(6.17 \text{ x } 10^{-3}) = 3.877 \text{ m}^2\text{/m}$

A_t = Total area per unit length = $A_f + \pi d(1\text{-}nb) = 3.877+ \pi\text{x}0.090\text{x}(1\text{-}200\text{x } 0.003)$

$= 3.877 +0.113 = 3.99 \approx 4.0 \text{ m}^2\text{/m}$

And $m = [2h_c/k_m/b]^{0.5}$ where h_c = thermal coefficient of outside pipe =

and k_m = thermal conductivity of fin = 21.4 w/mk

$m = 197.8$

Adopting fin efficiency from kays and London[43], $\eta = 0.35$

Overall heat Transfer coefficient U for finned portion of the heat pipe exposed to steam

$1/U = A_t/h_iA_i + (ff_i \text{ x } A_t/A_i) + ff_o+[(A_t/A_w \text{ x } d/2k_m)\text{x}\ln(d/d_i)]+ 1/\eta h_c$

A_t = Total external surface area of finned tube per unit length m^2/m = 4.0

A_i = the tube inner surface area per unit length = π d_i , m^2/m =0.251

A_w= average wall surface = π $(d+d_i)/2$, m^2/m = 0.267

K_m = thermal conductivity of tube wall = 21.4 w/mk

d,d_i = outer and inner diameters of m , 0.090,0.080

ff_i ,ff_o are the fouling factors inside and outside tubes m^2 k/w

h_i,,h_c are the tube inside and outside heat transfer coefficients, w/m^{2}k ,3724.7 kw/m^2.k, 1255.83 w/m^{2}k

η is the fin efficiency ,0.35

Neglecting the fouling factors,

$1/U = A_t/h_iA_{i\,+} + [(A_t/A_w \text{ x } d/2k_m)\text{xln}(d/d_i)]+ 1/\eta h_c$

$= [4/3724.7 \text{ x } 1000 \text{ x } 0.251] + [(4/0.267) \text{ x } (0.090/2\text{x}21.4)\text{x } \ln(0.090/0.080)] + 1/(0.35\text{x}1255.83)$

$= 4.278\text{x } 10^{-6} + [14.98\text{x}2.1\text{x } 10^{-3}\text{x}0.118]+ 2.275\text{x}10^{-3}$

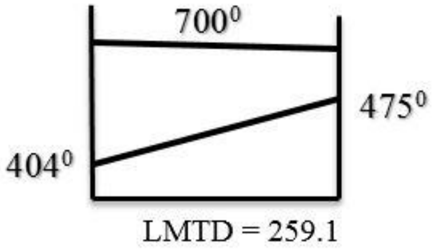
$= 5.99 \text{ x } 10^{-3} = 6 \text{ x } 10^{-3}$ w/m^{2}k

Hence, U = 166.7 w/m^{2}k

Calculations of Number of heat pipes required for the header duty

In the Steam Section:

Let N of heat pipes required for the load of 45 MW



$45\text{x } 10^6 = 166.7 \text{ x } 259.1 \text{ x } 4\text{x N}$ implies that N = 260.46 or 261 tubes

In the Flue gas Section:

h_1 = 262.5 w/m^2.k , h_2 = 853.13 kw/m^{2}K

A_1 = 3.4 m^2 A_2 = 3.0 m^2

$R_w = \ln (d/d_i) / (2\pi k_m L) = \ln(.090/.080)/ (2\pi \text{ x } 21.4\text{x } 12) = 7.3 \text{ x } 10^{-5}$

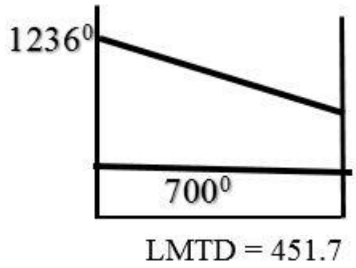
If overall coefficient of heat transfer is U, then

$1/U(3.4) = [1/(262.5)3.4] + 7.3 \text{ x } 10^{-5} + [1/853.13\text{x}1000)\text{x}3.0]$

$= 1.12 \text{ x } 10^{-3} + 7.3\text{x } 10^{-5} + 3.91\text{x}10^{-7} = 1.2 \text{ x } 10^{-3}$

U = 245.1 w/m^{2}k

Let N_1 of heat pipes required for the load of 45 MW



$45x\ 10^6 = 245.1 \times 451.7 \times 3.4 \times N_1$ implies that $N_1 = 119.5$ or 120 tubes

Pressure Drop across the finned tube bundle that is in the steam

$\Delta P_s = [0.205(f+a)G^2N_d]/\rho_{steam}$

N_d = Now rows in deep, in the direction of steam flow = 58

ρ_{steam} = Density of steam at average temperature ,kg/m^3 = 1/0.0172329 = 58.03

G = 393.6 kg/m^2.s

Friction factor = f = $C_2C_4C_6[(d+2h)/d]\ \{T_g/T_f\}^{0.25}$

$= 0.115 \times 0.509 \times 1.498 \times [(0.090+2 \times 0.025)/0.090]\ (713/873)^{0.25}$

$= 0.0877 \times 1.555 \times 0.951$

$= 0.1297$

$a=(1+B^2)(t_{g2}-t_{g1})/\{4N_d(t_g+273)\}$

B = Free steam flow area/Total flow area = $[(S_T - A_0)/S_T]^2$

$= [(0.19-0.12)/0.19]^2 = 0.136$

$a=(1+0.136^2)(475-404)/\{4 \times 58 \times (712.5)\} = 72.31/165300 = 4.37 \times 10^{-4}$

$\Delta P_s = [0.205(0.1297+4.37 \times 10^{-4})\ (393.6)^2 \times 58]/58.03$

= 239713.68/58.03 = 4130.86 mmWC

$= 4130.86 \times 9.8 \times 10^{-5}$ bar = 0.405 bar

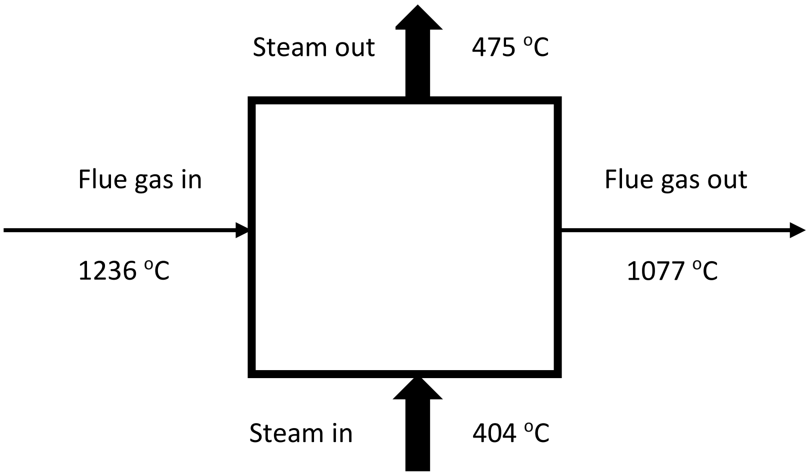
Whereas, in the conventional header the pressure drop is 3 bar

In the same lines the design of heat pipes for Final super heater and re heater applications are designed.

The exergy calculations for Platen super heater, Final super heater and Re heater Before and after proposed modifications with heat pipes

i. Platen Super Heater (PSH)

The inlet and out let temperature and pressure parameters for the PSH are as below.



Mass of steam handled = 165.3 kg/s

Mass of flue gas handled = 283.2 kg/s

Before modifications

Flue gas side

For flue gas stream, inlet exergy, $Ǝ_{fg,in} = m_{fg}$ x C_p x $[(T-T_o)- T_o \ln T/T_o]$

$= 283.2$ x $1.34 [(1236-25) – 298$ x$\ln (1236+273)/298]$

$= 276.1$ MW

For flue gas stream, outlet exergy, $Ǝ_{fg,out} = m_{fg}$ x C_p x $[(T-T_o)- T_o \ln T/T_o]$

$= 283.2$ x 1.3x $[(1077-25) – \ln (1077+273)/298]$

$= 221.5$ MW

Hence, Exergy input by flue gas = 276.1-221.5 = 54.6 MW

Steam side

Exergy gained by steam = $m_s[(h_{out} - h_{in}) - T_o (S_{in} - S_{out})]$

	Temp in C	Pressure in bar	Enthalpy , kJ/kg	Entropy kJ/kgK
Inlet	404	154	2981.4	5.88
outlet	475	151	3235.1	6.24

Exergy gained by steam = 165.3 [(93235.1-2981.4)-298(6.24-5.88)]

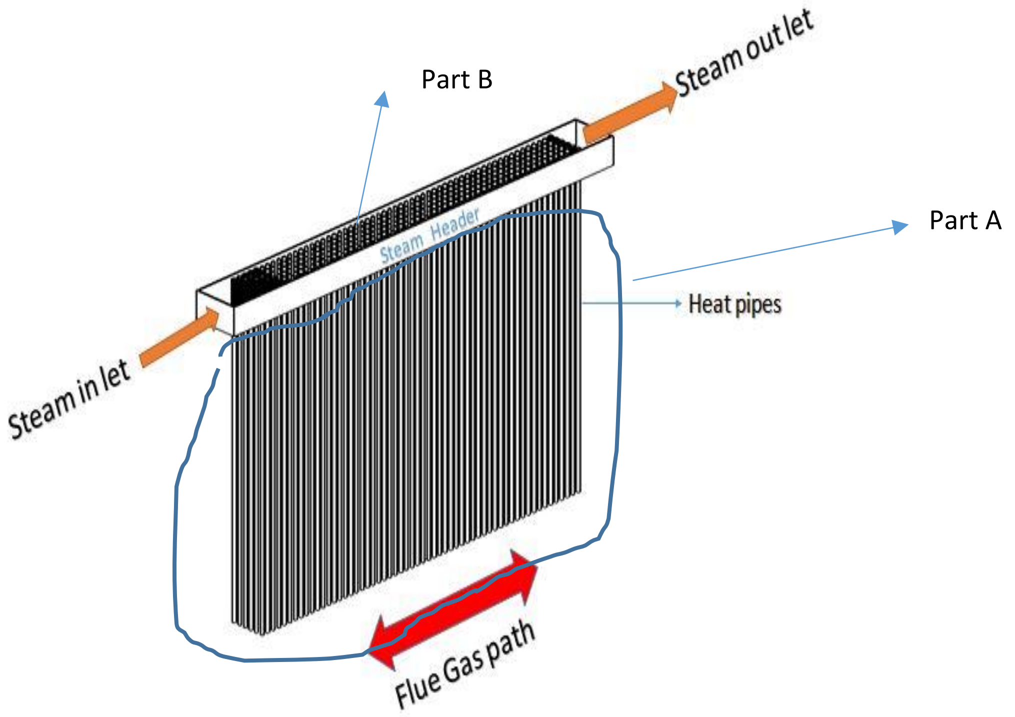
= 24.20 MW

Hence exergy destruction in PSH = Exergy input by flue gas - Exergy gained by steam

= 54.6-24.2

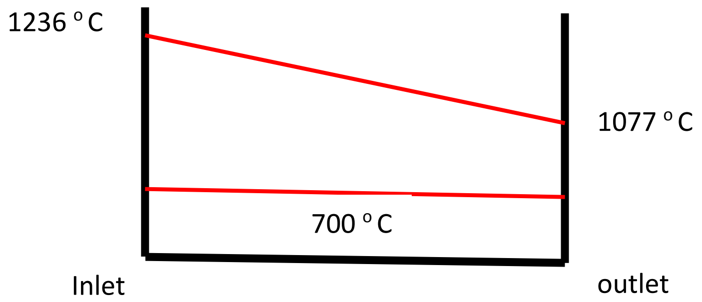
= **30.4 MW**

After Proposed modifications



<u>Part A (Flue gas section)</u>

In this section flue gas will get cooled (no phase change). Sodium liquid inside heat pipe gets converted into vapour, that is phase change occurs.



C_g = Heat capacity of flue gas = mass of the flue gas x Average specific heat of flue gas

= 283.2 [(1.34+1.3)/2] = 373.824

Exergy destruction in Part A (Flue gas Path)

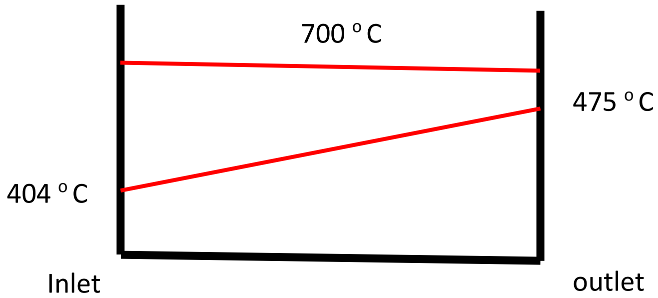
$$= T_o \times C_g \times [\ln(T_{out}/T_{in}) + (T_{in} - T_{out})/T_{cont}]$$

= 298x 373.824 [ln (1350/1509) + 159/973]

= 5.8 MW

<u>Part B (Steam section)</u>

In this section steam will get heated (no phase change). Sodium liquid inside heat pipe gets converted into liquid phase, that is phase change occurs.



Cs = Heat capacity of steam = mass of the flue gas x Average specific heat of steam

= 165.3 [(3.05+4.1)/2] = 590.9

Exergy destruction in Part B (Steam Path)

$$= T_o \times C_g \times [\ln(T_{out}/T_{in}) + (T_{in} - T_{out})/T_{cont}]$$

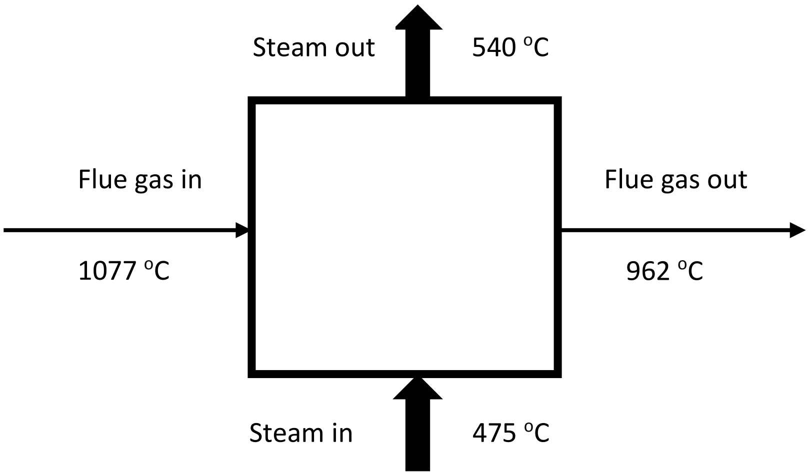
= 298x 590.9 [ln (748/677) + -71/973]

= 4.7 MW

Total Exergy destruction in PSH = 5.8 + 4.7 = 10.4 MW

ii. Final Super Heater (FSH)

The inlet and out let temperature and pressure parameters for the PSH are as below.



Mass of steam handled = 165.3 kg/s

Mass of Flue gas = 283.2 kg/s

<u>Before modifications</u>

<u>Flue gas side</u>

For flue gas stream, inlet exergy, $\text{Ɐ}_{\mathbf{fg,in}} = m_{fg} \times C_p \times [(T-T_o) - T_o \ln T/T_o]$

$= 283.2 \times 1.34 [(1077-25) - 298 \times \ln (1077+273)/298]$

$= 221.5$ MW

For flue gas stream, outlet exergy, $\text{Ɐ}_{\mathbf{fg,out}} = m_{fg} \times C_p \times [(T-T_o) - T_o \ln T/T_o]$

$= 283.2 \times 1.3 \times [(962-25) - 298 \times \ln (962+273)/298]$

$= 187.5$ MW

Hence, Exergy input by flue gas = 221.5-187.5 = 34.0 MW

<u>Steam side</u>

Exergy gained by steam = $m_s[(h_{out} - h_{in}) - T_o (S_{in} - S_{out})]$

Steam handled =

	Temp in C	Pressure in bar	Enthalpy , kJ/kg	Entropy kJ/kgK
Inlet	475	151	3235.1	6.24
outlet	540	148	3425.4	6.49

Exergy gained by steam = 165.3 [(3425.4-3235.1)-298(6.49-6.24)]

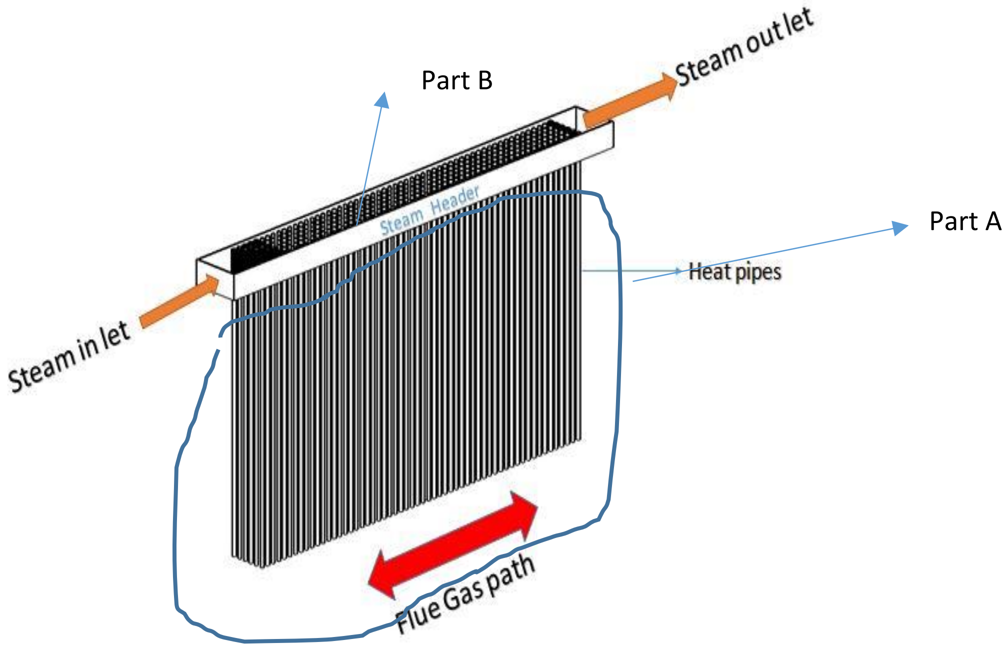
= 19.1 MW

Hence exergy destruction in FSH = Exergy input by flue gas - Exergy gained by steam

= 34.0-19.1

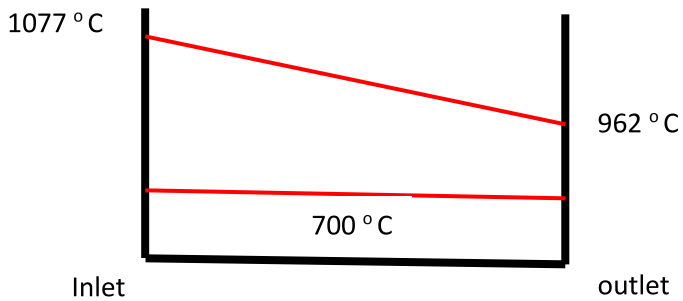
= **14.9 MW**

After Proposed modifications



Part A (Flue gas section)

In this section flue gas will get cooled (no phase change). Sodium liquid inside heat pipe gets converted into vapour, that is phase change occurs.



C_g = Heat capacity of flue gas = mass of the flue gas x Average specific heat of flue gas

= 283.2 [(1.34+1.29)/2] = 366.7

Exergy destruction in Part A (Flue gas Path)

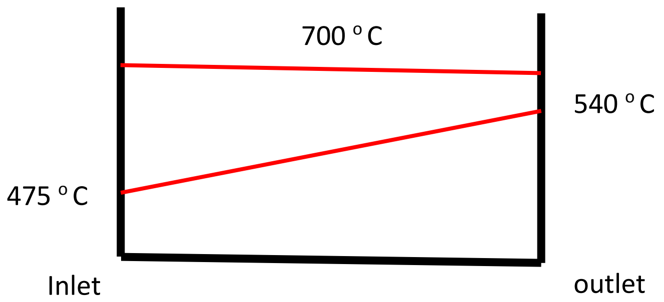
$$= T_o \times C_g \times [\ln(T_{out}/T_{in}) + (T_{in} - T_{out})/T_{cont}]$$

= 298x 366.7 [ln (1235/1350) + 115/973]

= 3.28 MW

__Part B (Steam section)__

In this section steam will get heated (no phase change). Sodium liquid inside heat pipe gets converted into liquid phase, that is phase change occurs.



Cs = Heat capacity of steam = mass of the flue gas x Average specific heat of steam

= 165.3 [(2.73+3.05)/2] = 477.7

Exergy destruction in Part B (Steam Path)

$= T_o \times C_g \times [\ln(T_{out} / T_{in}) + (T_{in} - T_{out})/T_{cont}]$

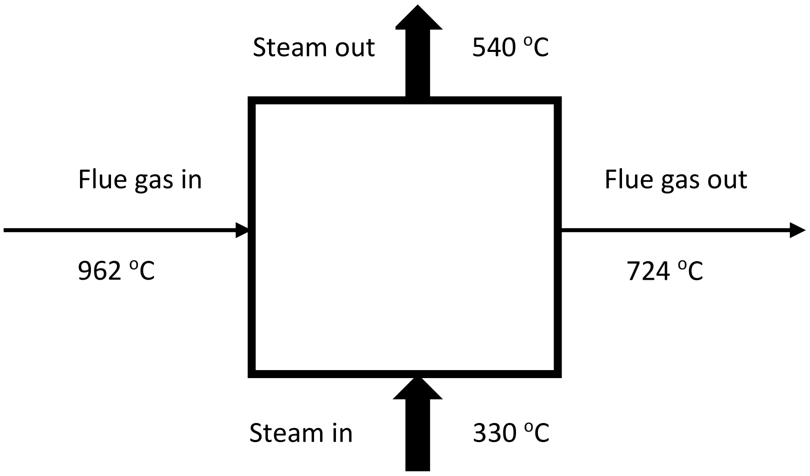
= 298x 477.7 [ln (813/748) + -65/973]

= 2.3 MW

Total Exergy destruction in FSH = 3.28 + 2.3 = 5.58 MW

iii. Re heater

The inlet and out let temperature and pressure parameters for the PSH are as below.



Mass of steam handled = 153.7 kg/s

Mass of Flue gas = 283.2 kg/s

Before modifications

Flue gas side

For flue gas stream, inlet exergy, $Ǝ_{fg,in} = m_{fg} \times C_p \times [(T-T_o)- T_o \ln T/T_o]$

$= 283.2 \times 1.29 [(962-25) - 298 \times \ln (962+273)/298]$

$= 188.8$ MW

For flue gas stream, outlet exergy, $Ǝ_{fg,out} = m_{fg} \times C_p \times [(T-T_o)- T_o \ln T/T_o]$

$= 283.2 \times 1.23 \times [(724-25) - 298 \times \ln (724+273)/298]$

$= 118.1$ MW

Hence, Exergy input by flue gas = 188.8-118.1 = 70.7 MW

Steam side

Exergy gained by steam = $m_s[(h_{out} - h_{in}) - T_o (S_{in} - S_{out})]$

Steam handled =

	Temp in C	Pressure in bar	Enthalpy , kJ/kg	Entropy kJ/kgK
Inlet	330	40	3042.5	6.5
outlet	540	37	3540.3	7.3

Exergy gained by steam = 153.7 [(3540.3-3042.5)-298(7.3-6.5)]

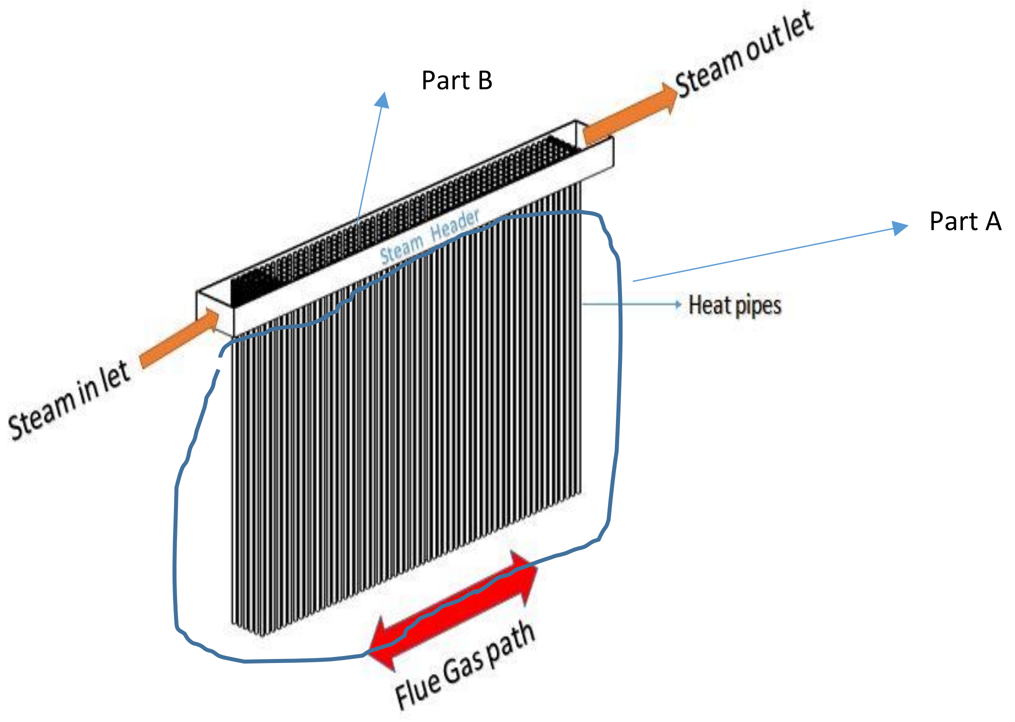
= 39.8 MW

Hence exergy destruction in FSH = Exergy input by flue gas - Exergy gained by steam

= 70.7-39.8

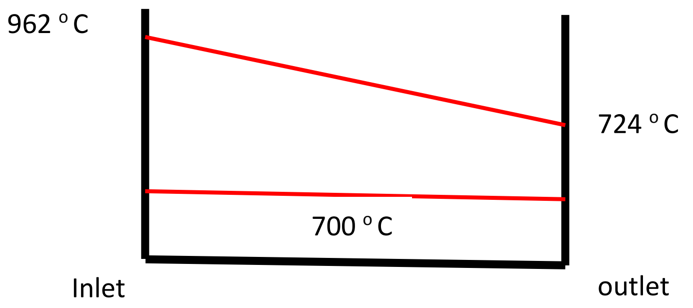
= **30.9 MW**

After Proposed modifications



Part A (Flue gas section)

In this section flue gas will get cooled (no phase change). Sodium liquid inside heat pipe gets converted into vapour, that is phase change occurs.



C_g = Heat capacity of flue gas = mass of the flue gas x Average specific heat of flue gas

= 283.2 [(1.23+1.299)/2] = 358.1

Exergy destruction in Part A (Flue gas Path)

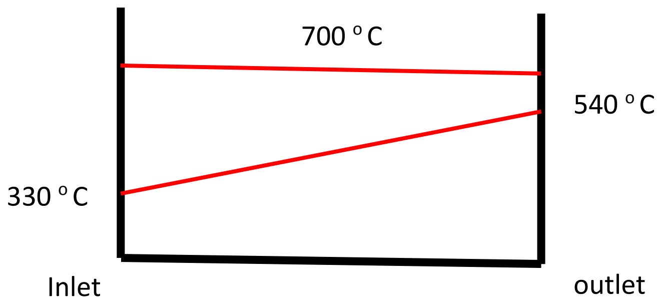
$$= T_o \times C_g \times [\ln(T_{out} / T_{in}) + (T_{in} - T_{out})/T_{cont}]$$

$$= 298 \times 358.1 [\ln(997/1235) + 238/973]$$

$$= 3.3 \text{ MW}$$

Part B (Steam section)

In this section steam will get heated (no phase change). Sodium liquid inside heat pipe gets converted into liquid phase, that is phase change occurs.



Cs = Heat capacity of steam = mass of the flue gas x Average specific heat of steam

$$= 153.7 [(2.6+1.3)/2] = 300$$

Exergy destruction in Part B (Steam Path)

$$= T_o \times C_g \times [\ln(T_{out} / T_{in}) + (T_{in} - T_{out})/T_{cont}]$$

$$= 298 \times 300 [\ln(813/603) + -210/973]$$

$$= 7.4 \text{ MW}$$

Total Exergy destruction in Re heater = 3.3+ 7.4 = 10.7 MW

Appendix IX

5E calculations for the case study with Proposed Modifications

A. *After modifications with Heat pipes*

i. Efficiency (η)

η = Energy output / energy input

= (Energy out – Auxiliary consumption) / Calorific value of coal x coal consumption

= [(191-8.60) x 1000] / 14654.5 x 40

= 31.12 %

ii. Exergy efficiency (Ǝη)

$Ǝ_{d,b}$ = Exergy desturction in the boiler = 373.32 MW

$Ǝ_{d,T}$ = Exergy desturction in the turbine = 16.10 MW

$Ǝ_{d,c}$ = Exergy desturction in the condenser = 5.9 MW

$Ǝ_{\eta,u}$ = 1- (Exergy Destruction/Exergy input)

= 1- (373.32+16.10+5.9)/850.2

= 1- 0.4649 = 53.5 %

iii. Exergoeconomic Factor (ϒ)

$ϒ = Ǝ_{out} (Ǝ_{\eta\ 1}/1-Ǝ_{\eta\ 1}) = 477.7 \{ 0..562/(1-0.562) \} = 616.2$

iv. Exergy Renewability index (ƎRI)

$Ǝ_{RI}$ = 191/ (621.3+372.66+6.8) = 19.1 %

v. Endurability factor (Ə)

Ə = Exergy destruction / Exergy input

= (350.48+16.10+5.9)/621.3 = 0.599

B. After modifications with Heat pipes and Solar steam blending

vi. Efficiency (η)

η = Energy output / energy input

= (Energy out – Auxiliary consumption) / (Calorific value of coal x coal consumption + Solar Energy input)

= [(191-8.60) x 1000] / (14654.5 x 32 +97740)

= 32.2 %

vii. Exergy efficiency ($Ǝ_\eta$)

$Ǝ_{d,b}$ = Exergy desturction in the boiler = 366.61 MW

$Ǝ_{d,T}$ = Exergy desturction in the turbine = 16.10 MW

$Ǝ_{d,c}$ = Exergy desturction in the condenser = 5.9 MW

$Ǝ_{\eta,u}$ = 1- (Exergy Destruction/Exergy input due to coal + Exergy input due to solar)

= 1- (366.61+16.10+5.9)/(816.71)

= 1- 0.4758 = 52.4 %

viii. Exergoeconomic Factor (Υ)

Υ = $Ǝ_{out}$ ($Ǝ_{\eta\ 1}$/1-$Ǝ_{\eta\ 1}$) = 477.7 { 0..66/(1-0.66) } = 927.3

ix. Exergy Renewability index ($Ǝ_{RI}$)

$Ǝ_{RI}$ = 191/ (497+248.3+6.8) = 25.4 %

x. Endurability factor (Ə)

Ə = Exergy destruction / Exergy input

= (226.3+16.10+5.9)/427 = 0.581

Appendix X

Solar Power Tower (SPT) PROPOSAL

The proposed Solar Tower system for the present application is depicted in Fig.A.8.

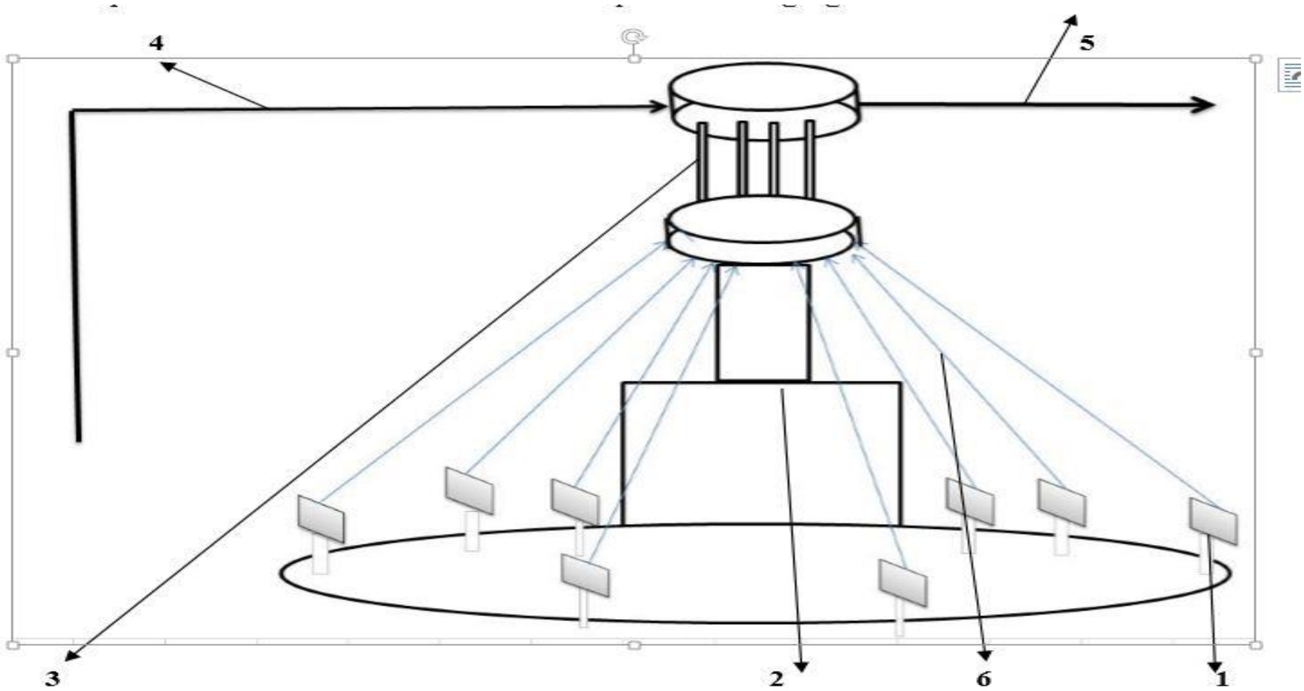


Fig A.8 : The Solar Tower and Heliostat Arrangement

The solar radiations are reflected from the heliostats (1) on the solar Tower (2) The heliostats are controlled by solar trackers, so that maximum reflections will resulted from the Sun. The solar tower receives the radiation from the reflectors and raises the temperature of the liquid it contains. The Heat pipes (3) that are present in the solar tower transmits the heat to the steam inlet (Received through pipe 4). The inlet temperature of the steam supplied by pipe 4 , which is economizer outlet is 300° C. After receiving the heat through the heat pipes the outlet temperature of steam will be around 550°C. This steam can be supplied to the utilization point through the pipe 5. In India, this solar hybridization may be utilized during 8.30 AM to 4.30 PM of the day during which maximum solar radiations available in India. Rest of the day the conventional coal can be utilized. The Figure A.9 presents a pictorial representation of the receiver.

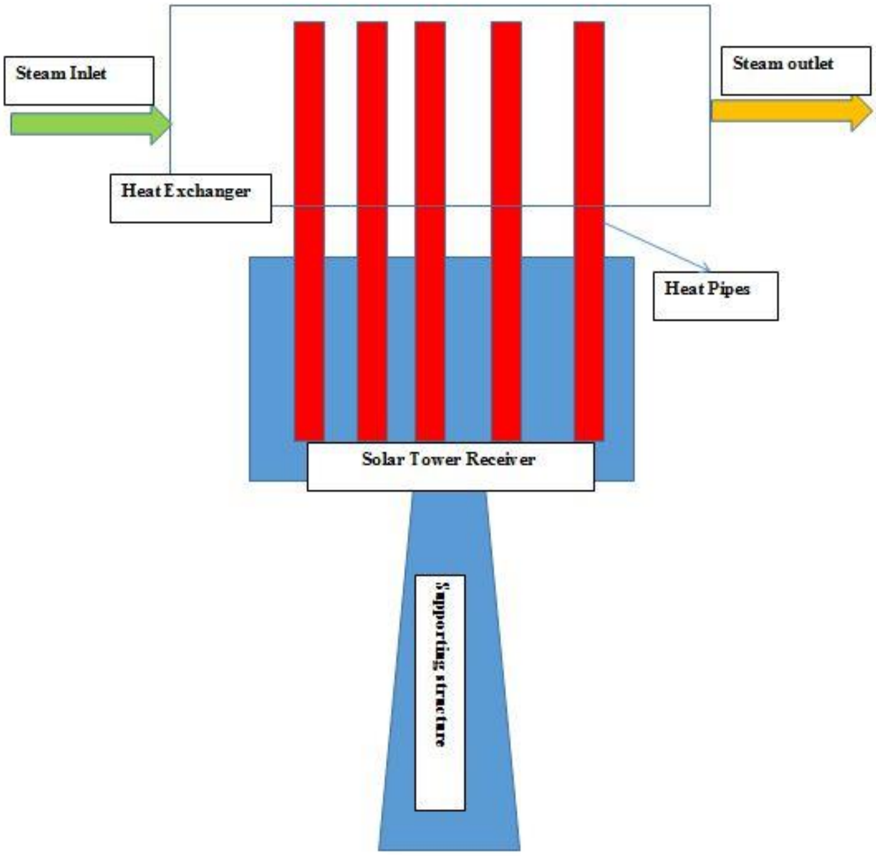


Fig A.9. Outline of the Solar Tower Receiver

Appendix XI

EXPERIMENTAL RESULTS

TABLE A5. EXPERIMENTAL RESULTS OF SINGLE HEAT PIPE WITH HOT WATER

Experiment No	**Hot Fluid temp in ºC**				**Cold Fluid temp in ºC**				**kW**		
									Energy supplied to Heat Pipe	**Energy Transferred by Heat Pipe**	**Difference or Un accountable energy loss**
	T_1	T_2	ΔT_H	Quantity in lit /min	T_3	T_4	ΔT_C	Quantity in lit/min			
1	66.4	56.0	10.4	6.5	26.8	36.9	10.1	6.5	4.72	4.57	0.15
2	57.4	47.4	10.0	6.5	26.8	35.5	8.7	7.0	4.54	4.25	0.29
3	56.8	45.5	11.3	5.8	22.3	32.5	10.2	6.2	4.57	4.41	0.16
4	56.5	45.5	11.0	6.1	22.4	32.5	10.1	6.5	4.68	4.41	0.27
5	56.3	45.5	10.8	6.0	22.3	32.5	10.2	6.0	4.52	4.27	0.25
6	54.9	44.0	10.9	6.0	26.5	37.0	10.5	6.1	4.56	4.30	0.26
7	49.8	39.1	10.7	6.3	22.8	32.5	9.7	6.6	4.70	4.47	0.23
8	49.4	38.0	11.4	6.0	22.8	32.0	9.2	7.1	4.78	4.56	0.22
9	43.0	31.9	11.1	6.2	22.6	32.0	9.4	6.8	4.8	4.46	0.34
10	42.1	32.0	10.1	6.5	22.6	32.1	9.5	6.8	4.72	4.50	0.22

TABLE A6. EXPERIMENTAL RESULTS OF SINGLE HEAT PIPE WITH STEAM

Sl. No	$\dot{M}$ condensate Lit/min	T_1 of Condensate in °C	T_2 of steam inlet in °C	P of steam inlet in bar	$\dot{m}$ of cooling water in lit/min	T_3 of inlet cooling water °C	T_4 of outlet cooling water °C	Q_1 heat input of condensate in kW	Q_2 heat energy carried out by condensate in kW	Q_3 Heat output of cooling water in kW	Difference of heat input and output in kW $\Delta Q = (Q_1-\{Q_2+Q_3\})$
1	0.760	42.5	44	0.09	33	22.5	32.0	29.659	2.32	21.84	5.50
2	0.762	43.0	44	0.09	33	22.5	32.3	29.810	2.34	22.53	4.94
3	0.769	43.5	44	0.09	33	22.5	32.3	30.096	2.36	22.53	5.21
4	0.775	43.8	44	0.09	33	22.5	33.0	30.360	2.37	24.14	3.85
5	0.782	45.3	46	0.10	35	22.5	32.8	30.734	2.50	25.11	3.13
6	0.789	47.5	48	0.11	35	22.5	33.0	31.020	2.64	25.60	2.78
7	0.792	49.4	50	0.12	35	21.0	31.5	31.306	2.76	25.60	2.95
8	0.794	51.6	52	0.13	35	21.0	32.0	31.592	2.87	26.82	1.90
9	0.796	52.6	53	0.14	35	21.0	31.5	31.700	2.92	25.60	3.18
10	0.800	52.8	53	0.14	35	21.0	32.0	31.850	2.93	26.82	2.10

TABLE A 7. EXPERIMENTAL RESULTS OF HEAT PIPE CONDENSER

Sl. No	$\dot{M}_{condensate}$ Lit/min	T_1 of Condensate in °C	T_2 of steam inlet in °C	P of steam inlet in bar	$\dot{m}$ of cooling water in lit/min	T_3 of inlet cooling water °C	T_4 of outlet cooling water °C	Q_1 heat input of condensate in kW	Q_2 heat energy carried out by condensate in kW	Q_3 Heat output of cooling water in kW	Difference of heat input and output in kW $\Delta Q = (Q_1 - \{Q_2 + Q_3\})$
1	0.18	61.5	62.0	0.22	35	26.5	30.5	11.73	0.78	9.75	1.2
2	0.20	65.8	66.0	0.26	35	26.5	30.8	11.80	0.92	10.48	0.4
3	0.22	75.5	76.0	0.40	35	26.5	30.8	12.00	1.16	10.48	0.36
4	0.85	46.0	46.5	0.10	35	26.0	36.3	30.734	2.72	25.12	2.89
5	0.89	47.5	48.0	0.11	33	26.0	38.1	30.800	2.96	27.82	0.02
6	0.90	49.4	50.0	0.12	35	26.0	37.0	31.306	3.10	26.82	1.39
7	0.96	51.0	52.0	0.13	33	26.0	37.8	31.592	3.42	27.13	1.04
8	1.05	51.6	52.0	0.13	35	26.0	37.2	31.600	3.74	27.31	0.55
9	1.00	52.6	53.0	0.14	35	26.0	37.0	31.700	3.52	26.82	1.36
10	1.08	52.8	53.0	0.14	33	26.0	38.0	31.850	3.96	27.58	0.31

Appendix XII

Error Analysis for Table A 5

TABLE A 8. UNCERTAINTY ANALYSIS OF TEST PARAMETERS AND EQUIPMENT

Measurement instrument/property	**Working Range of the Device**	**Measured Property**	**Test Range**	**Device Measurement error**
Digital Thermometer 1	-50 to 300^o C	Hot Fluid Temp	40-75^o C	0.7
Digital Thermometer 2	-50 to 300^o C	Hot Fluid Temp	30-60^o C	0.7
Digital Thermometer 3	-50 to 300^o C	Cold Fluid Temp	20-30^o C	0.71
Digital Thermometer 4	-50 to 300^o C	Cold Fluid Temp	20-40^o C	0.71
Flow meter 1	0.5- 25 l/min	Hot Fluid Quantity	5- 10 l/min	0.6
Flow meter 2	0.5- 25 l/min	cold Fluid Quantity	5- 10 l/min	0.6

The parameter uncertainty errors associated with the experimental results are as below

Energy supplied = $mc_p\Delta T = \{ \sum_{j=1,2}(m_j/\partial m)^2 + \sum_{j=1-4}(T_j/\partial T)^2 \}^{0.5}$ < 5%

Energy delivered = $mc_p\Delta T = \{ \sum_{j=1,2}(m_j/\partial m)^2 + \sum_{j=1-4}(T_j/\partial T)^2 \}^{0.5}$ < 4.8%

Error Analysis for Table A 6

TABLE A 9. UNCERTAINTY ANALYSIS OF TEST PARAMETERS AND EQUIPMENT

Measurement instrument/property	**Working Range of the Device**	**Measured Property**	**Test Range**	**Device Measurement error**
Digital Thermometer 1	-50 to 300^o C	Hot Fluid Temp	44-55^o C	0.7
Digital Thermometer 2	-50 to 300^o C	Hot Fluid Temp	40 -60^o C	0.7
Digital Thermometer 3	-50 to 300^o C	Cold Fluid Temp	20-30^o C	0.71
Digital Thermometer 4	-50 to 300^o C	Cold Fluid Temp	20-40^o C	0.71
Flow meter 1	0.5- 25 l/min	Hot Fluid Quantity	5- 10 l/min	0.6
Voltmeter	0-300 V	voltage	220-230 V	0.1
Ammeter	0-300 A	Amperage	100-200 A	0.1

The parameter uncertainty errors associated with the experimental results are as below

Energy supplied = VI x power factor = $\{ \sum_{j=1,2}(v_j/\partial v)^2 + \sum_{j=1\text{-}4}(I_j/\partial I)^2 \}^{0.5}$ < 2.7%

Energy delivered = $mc_p\Delta T$ = $\{ \sum_{j=1,2}(m_j/\partial m)^2 + \sum_{j=1\text{-}4}(T_j/\partial T)^2 \}^{0.5}$ < 6.0%

Error Analysis for Table A 7

TABLE A 9. UNCERTAINTY ANALYSIS OF TEST PARAMETERS AND EQUIPMENT

Measurement instrument/property	**Working Range of the Device**	**Measured Property**	**Test Range**	**Device Measurement error**
Digital Thermometer 1	-50 to 300° C	Hot Fluid Temp	44-80° C	0.7
Digital Thermometer 2	-50 to 300° C	Hot Fluid Temp	44-80° C	0.7
Digital Thermometer 3	-50 to 300° C	Cold Fluid Temp	20-30° C	0.71
Digital Thermometer 4	-50 to 300° C	Cold Fluid Temp	20-40° C	0.71
Flow meter 1	0.5- 25 l/min	Hot Fluid Quantity	5- 10 l/min	0.6
Voltmeter	0-300 V	voltage	220-230 V	0.1
Ammeter	0-300 A	Amperage	100-200 A	0.1

The parameter uncertainty errors associated with the experimental results are as below

Energy supplied = VI x power factor = $\{ \sum_{j=1,2}(v_j/\partial v)^2 + \sum_{j=1\text{-}4}(I_j/\partial I)^2 \}^{0.5}$ **< 2.7%**

Energy delivered = $mc_p\Delta T = \{ \sum_{j=1,2}(m_j/\partial m)^2 + \sum_{j=1\text{-}4}(T_j/\partial T)^2 \}^{0.5}$ **< 6.5%**