Parametric Analysis and Optimization of Heat Exchangers in a Thermal Power Plant

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Major Project

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(Thermal Engineering)

By

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Declaration

This is to certify that

1. The thesis comprises of my original work towards the degree of Master of Technology in Mechanical Engineering (Thermal Engineering) at Nirma University and has not been submitted elsewhere for a degree.

2. Due acknowledgment has been made in the text to all other material used.

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Abstract

Heat Exchangers are equipments targeted to the efficient transfer of heat from a hot fluid to a cold fluid, mostly through an intermediate metallic wall and without moving parts. The Heat Exchangers are one of the important and critical equipments for the power generation technology as the overall performance of a power plant system is directly affected by the heat exchanger's performance. For a large capacity power plant, a small change in the output makes a huge difference in considerations such as performance effectiveness, total cost of equipment etc. In the present study, an investigation is carried out for the design and performance analysis of heat exchangers such as surface condenser and feedwater heaters used in power plants by changing various critical parameters and optimization of those parameters for power plant optimization. The HEI (Heat Exchanger Institute) Standard's method is used for designing the heat exchangers and calculations using excel functions is carried out for the thermal design of heat exchangers.

The present study is conducted considering parameters such as TTD (Terminal Temperature Difference), Tube Outer Diameter, Tube Material and Cooling water Velocity for a surface condenser. The present study indicates that in case the value of TTD is increased, while keeping the heat duty and length of tube constant, all other parameters such as the number of tubes, the mass flow rate of cooling fluid and the effective heat transfer area decreases. The thermal design calculations considering SS439 as a tube material (instead of SS304) reveal that there is a saving of about 10% in the overall weight of tube material utilized. For the feedwater heaters, the present study is conducted considering TTD analysis and Desuperheating Zone addition in the LP heater. The present study indicates that in case the value of TTD increases, the total area of heater linearly decreases in the LP heater whereas the total area of heater exponentially decreases in the HP heater. A comparison of LP heater without Desuperheating Zone and with Desuperheating Zone shows that if the superheat is available with steam and a Desuperheating Zone is provided to LP heater then the total area of LP heater decreases at a remarkable level as compared to the total area of LP heater without a Desuperheating Zone.

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Nomenclature

η	Cycle Efficiency
Q	Heat Duty , kCal/hr
h	Enthalpy , kJ/Kg
Wp	Work of Pump , kJ/s
m	Mass Flow Rate , kg/s
C_p	Specific Heat , kJ/kg K
TR	Temperature Rise , °C
T_1	Cooling water inlet Temperature , °C
T_2	Cooling water outlet Temperature , °C
T_s	Saturation Temperature , °C
V	Specific Volume , m^3/kg
U	Overall Heat Transfer Coefficient , W/m^2 K
U_1	Uncorrected Overall Heat Transfer Coefficient , W/m^2 K
F_m	Material Correction Factor
F_c	Cleanliness Factor
F_w	Inlet Cooling water Temperature Correction Factor
LMTD	Log Mean Temperature Difference , °C
A	Effective Heat Transfer Area , m^2
N_t	Number of Tubes
R_1	Temperature Factor
R_{iw}	Waterbox Inlet Loss
R_2	Tube Outer Diameter and Gauge Factor
R_{ow}	Waterbox Outlet Loss
R_t	Friction Loss
R_{te}	Tube Loss (Inlet + Outlet)
r	Density of fluid , kg/m^3
u	Velocity of fluid , m/s
A_c	Cross-section Area , m^2
d_i	Tube inside diameter , mm
Re	Reynolds number
m	Dynamic Viscosity , kg/m s
Pr	Prandtl number
f	Friction factor
Nu	Nusselt number
CL	Tube Layout Constant
CTP	Tube Count Calculation Constant
D_e	Shell equivant Diameter
C	Clearance , mm
B	Baffle Spacing , m
U_f	Fouling Heat Transfer Coefficient , W/m^2 K
h_{fg}	Latent heat , kJ/kg
T_{film}	Film Temperature , °C
h_{cond}	Condensing Heat Transfer Coefficient , W/m^2 K
T_m	Tube Mean Metal Temperature , °C
r_i	Fouling resistance inside the surface of tubes , m^2 K/W
r_o	Fouling resistance outside the surface of tubes , m^2 K/W
r_w	Tube wall resistance , m^2 K/W

Chapter 1

Introduction

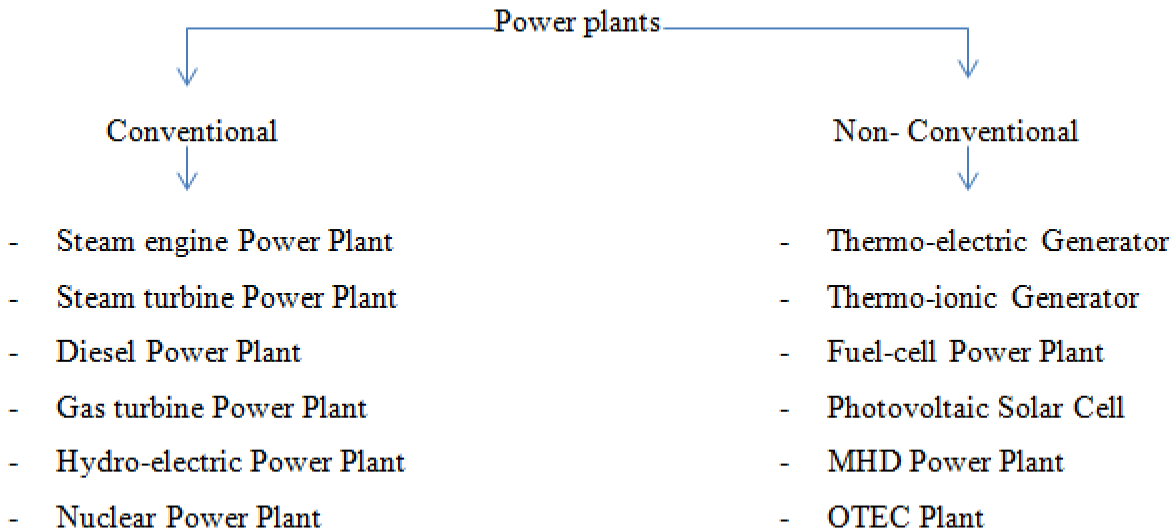
1.1 Introduction to Power Plants

Electricity is useful to us in our daily life and one of its major generating location is “the power plant”.

There are mainly two types of power plants.

1. Conventional
2. Non- Conventional

An overview of power plants is as following :



One of the major source of generating power is a thermal power plant. In a thermal power plant, the power is generated by driving a large electrical generator (also called a turbo-generator). This generator is driven by a shaft coupled turbine which rotates by using the force of steam. Hence in order to produce the electricity, our primary concern is to produce steam from water by using thermal energy. This thermal energy is produced by burning fossil fuels to produce a sufficient amount of heat to generate steam from water. In a thermal power plant, there are a large number of processes involved such as fuel burning, boiling of water, generation of steam, rotation of turbine, generation of electricity etc. These processes need a large number of parts, equipments and they are arranged in a definite manner[1]. An overview of a coal fired thermal power plant is shown in figure 1.1.

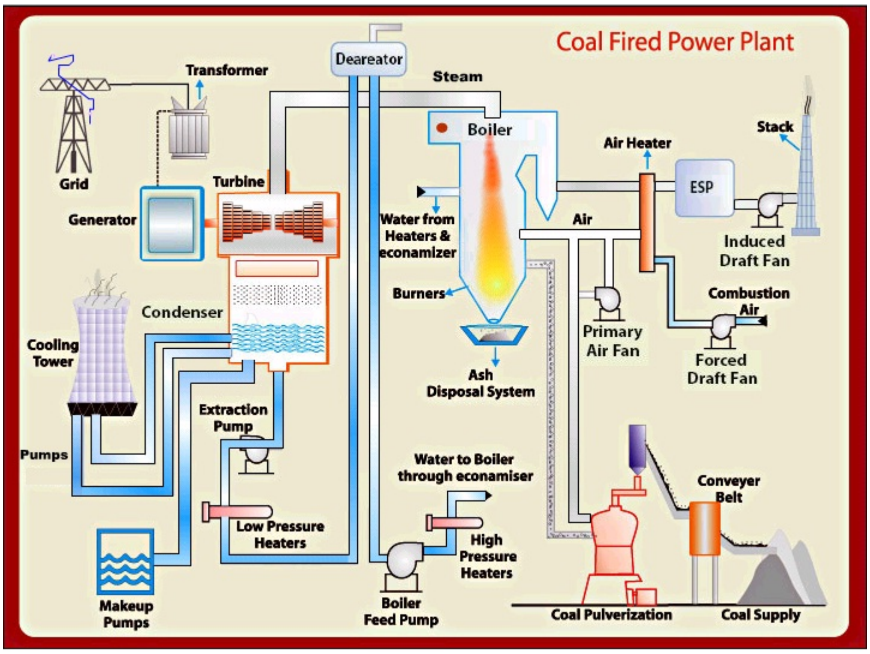


Figure 1.1: **Coal-fired Thermal Power Plant**[1]

1.2 Thermal Cycles

Power generation process is cyclic process and working fluid rotates in cycle during the power production. This cycle is known as the Thermal cycle[1].

The various thermal cycles include the following :

1. Carnot Cycle

2. Vapour Power Cycle

- Rankine Cycle
- Optimized Rankine Cycle

3. Gas Power Cycle

- Brayton Cycle
- Cogeneration Cycle

4. Combined Cycle

5. Super-critical Power Cycle

1.2.1 Carnot Cycle

The Carnot Cycle is a theoretical thermodynamic cycle proposed by Nicholas Leonard Sadi Carnot in 1824. The Carnot Cycle is an ideal simple cycle consisting of isentropic compression, isentropic expansion, isothermal heat addition and isothermal heat rejection. The Carnot cycle delivers the highest efficiency of Power Plant and it is not dependent on the fluid properties.

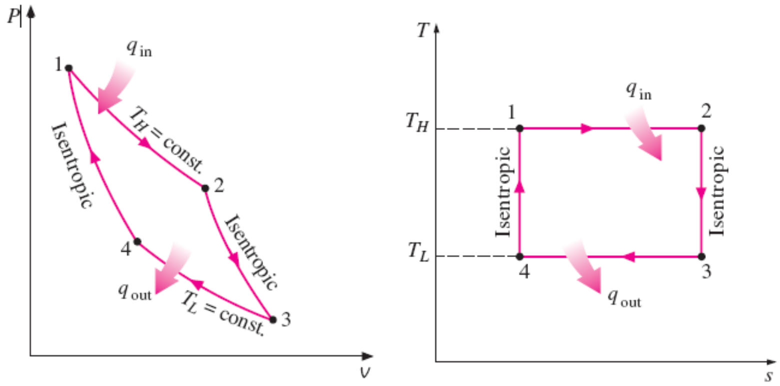


Figure 1.2: **P-V and T-S diagram of Carnot cycle**[7]

The four reversible proccesses of Carnot cycle are as folowing:

- 1 to 2: Reversible isothermal expansion.
- 2 to 3: Reversible adiabatic expansion
- 3 to 4: Reversible isothermal compression
- 4 to 1: Reversible adiabatic compression

The Carnot principles for heat engine are as given below:

- The efficiency of all reversible heat engines operating between a constant temperature source (T_H) and a constant temperature sink (T_L) are the same. They only depend on high temperature T_H = constant and low temperature T_L = constant.

- Between the same two reservoirs, the efficiency of reversible heat engine is always greater than the efficiency of an irreversible one.

1.2.2 Vapour Power Cycle

The steam cycle is a practical one and it is the basis of virtually all steam power plants and hence electricity generation. The main difficulties of the Carnot cycle are overcome by complete condensation in the condenser and by superheating (optional).

a) Rankine Cycle

The Rankine cycle is the basic working cycle of all power plants where the working fluid is consistently vaporized and condensed. The choice of working fluid depends on temperature range. In the Rankine cycle, steam flows to a turbine where part of its energy is changed over to mechanical energy. The reduced energy steam flowing out of the turbine condenses to water in the condenser.

A feed water pump gives back the condensed fluid (condensate) to the boiler. The rejected heat from the steam entering the condenser is exchanged to a different cooling water loop.

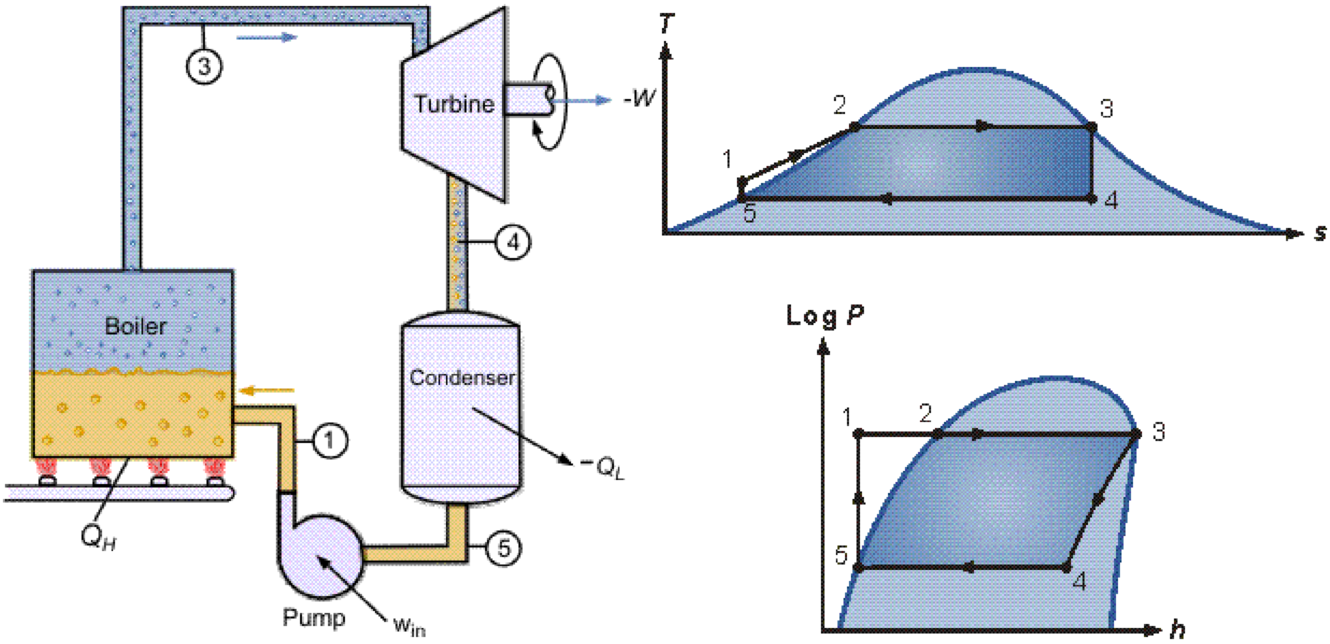


Figure 1.3: **Ideal Rankine cycle with T-S and P-h diagram[1]**

In the Rankine cycle, the steam flows to a turbine in which the part of steam energy is converted into the mechanical energy. Because of energy transformation, the reduced energy steam leaving turbine changed over to water by condensation process in the condenser.

A pump conveys the condensed fluid (water) back to the evaporator. The rejected heat from the steam in the condenser is exchanged to the cooling water loop set in the condenser separately.[2]

The Ideal Rankine Cycle does not include any internal irreversibility and comprises of following four procedures:

• 1 to 2: Water from the condenser at low pressure is pumped into boiler at high pressure. This process is reversible adiabatic.
• 2 to 3: Water is converted into steam at constant pressure by addition of heat in the boiler.
• 3 to 4: Reversible adiabatic expansion of steam in the turbine.
• 4 to 1: Constant pressure heat rejection in the condenser to convert condensate into water.

The thermal efficiency of the Rankine cycle can be given by following formula:

$$\eta = \frac{Q_1 - Q_2}{Q_1} \tag{1.1}$$

Where Q1 = Heat addition in Boiler

Q2 = Heat rejection from condenser

b) Optimized Rankine Cycle

Reheat Rankine cycle :

In simple Rankine cycle, after the isentropic expansion in turbine, the steam is directly fed into condenser for the condensation process. But in reheat system, two turbines (high pressure turbine and low pressure turbine) are utilized for improving efficiency. Steam after expansion from high pressure turbine is sent again to boiler and heated till it comes to superheated condition. It is then left to expand in low pressure turbine to achieve condenser pressure.

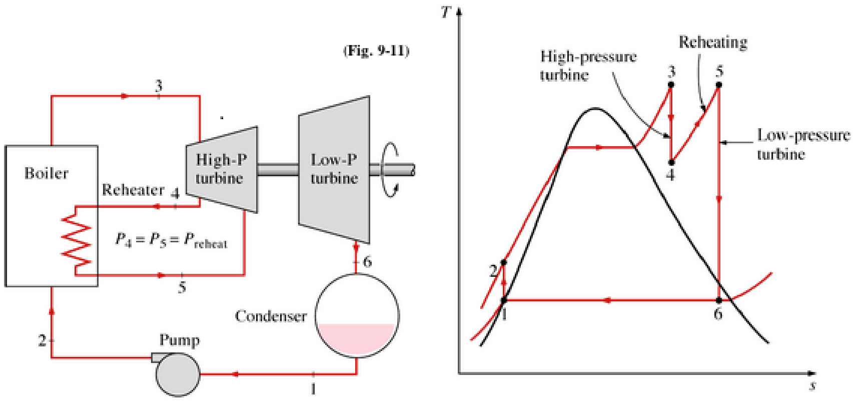


Figure 1.4: **Schematic diagram and T − S diagram of Reheat Rankine Cycle[1]**

It is obvious from the T-S graph that there is very little gain in efficiency from reheating the system, because the normal temperature at which heat is provided is not incredibly changed. In the event that metals could be found that would empower us to superheat the system to 3', the simple Rankine cycle would be more productive than the reheat cycle.

The efficiency of reheat Rankine cycle is given by following equation.

$$\eta = \frac{(h_3 - h_4) - (h_5 - h_6) - w_p}{h_3 - (h_1 + w_p) + (h_5 - h_4)} \tag{1.2}$$

Regenerative Feed Heating Cycle

From the T − S diagram of rankine cycle, it is evident that at state 2-2' the working fluid enters the boiler and this temperature is very low temperature at which water is entering the boiler. As a result of the efficiency of cycle is lower.

Regeneration :

There is an approach to overcome this issue by raising the temperature of the working fluid before it enters the boiler and this procedure is called regeneration in steam power plant.

Conventional way for doing regeneration in a power plant is by separating steam from the turbine after fractional expansion or partial work done. This steam is utilized to heat the feed water and the device in which it happens is known as a feedwater heater or regenerator.

Regeneration enhances the cycle productivity by increasing the initial feed water temperature before the water enter the boiler and furthermore helps in controlling the large flow rate of steam at the turbine exhaust.

Regeneration is generally utilized as a part of all power plants where productivity is of significance and fuel saving is a motto.

A feedwater heater is fundamentally a heat exchanger where heat is exchanged to the feed water by extracting the partially expanded steam from the turbine to heat the feed water. the heating of feedwater is possible by:

1) Direct Heating :

Direct heating of feed water is performed in tanks or vessel also called open feed water heaters.

2) Indirect Heating (Shell & Tube Heat Exchanger) :

Indirect heating of steam and water is performed on shell & tube type closed feed water heaters.

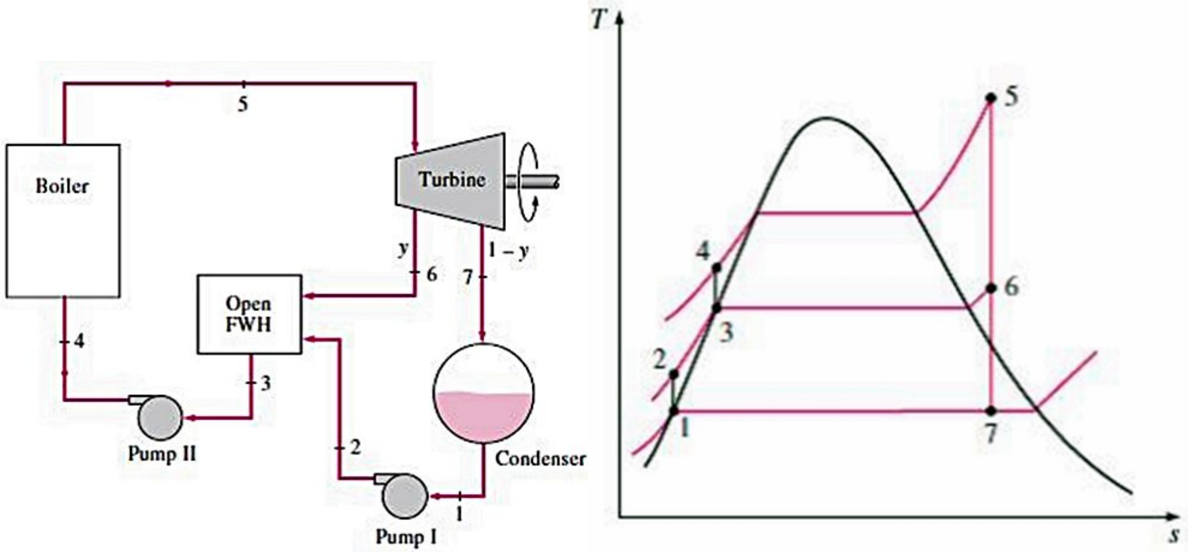


Figure 1.5: **Schematic diagram and T-S diagram of Regenerative Feedwater Heating Cycle**[1]

The greater number of feed heaters introduced the greater improvement in thermal efficiency. However, the incremental gain for each extra feed heater decreases as the number of heaters for a specific size of plant is computed on feasibility basis. The extra advantages of regenerative feedwater heating can be listed as follows :

- Increased steam flow in initial stages resulted in increased blade heights, which improves the internal efficiency of turbine.

• Reduced flow at turbine exhaust demands lesser exhaust area, resulting in smaller blades in last stages, which is the limiting factor in turbine design.

The decrease in steam flow at turbine exhaust additionally reduce the flow of working fluid in condenser, condensate pumps, ejectors and low-pressure heaters, thereby decreasing their sizes and leads to saving in capital investment.

1.2.3 Gas Power Cycle

a) Brayton Cycle

The air standard Brayton cycle is the ideal cycle for the simple gas turbine. It utilizes a single phase, gaseous fluid. The fig 1.6 shows the simple open cycle gas turbine.

The efficiency of the air-standard Brayton cycle is an element of the isentropic pressure ratio. The fact that the efficiency increases with the pressure ratio is evident from the T-s diagram. The cycle has a greater heat supply and a similar heat rejected as the original cycle.

The real gas turbine cycle varies from the ideal cycle principally as a result of irreversibility in the compressor and turbine, and as a result of pressure drops in the stream sections and ignition chamber (or in the heat exchanger of a closed- cycle turbine).

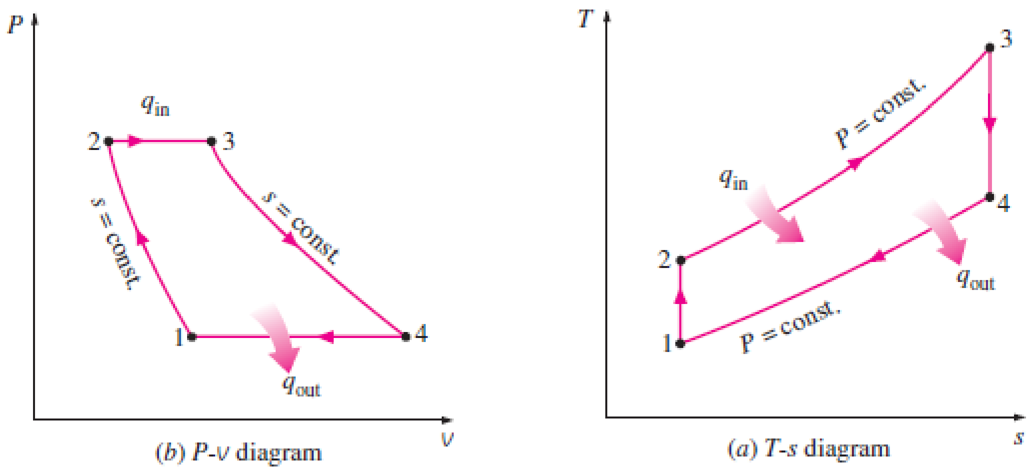


Figure 1.6: **P-V and T-S diagram of Brayton cycle[2]**

b) Cogeneration Cycle :

The Need for Cogeneration :

Thermal power plants are a major source of power supply in India. The regular strategy for power generation and supply to the client is inefficient as in just about 33% of the essential energy fed into the power plant is really made accessible to the client as power. In conventional power plant, efficiency is just 35% and remaining 65% of energy is lost. The significant source of loss in the conversion procedure is the heat rejected to the surrounding water or air because of the limitations of the distinctive thermodynamic cycles utilized in power generation. Likewise further losses of around 10-15% are related with the transmission and distribution of power in the electrical network.

The Principle of Cogeneration :

Cogeneration or Combined Heat and Power (CHP) are characterized as the consecutive generation of two distinct types of valuable energy from a single primary energy source, normally mechanical energy and thermal energy. Mechanical energy might be utilized to drive an alternator for producing power, or pivoting gear, for example, engine, compressor, pump or fan for conveying different services. The thermal energy can be utilized either for direct process applications or for an indirect producing steam, high temp water, hot air for dryer or chilled water for process cooling.

The general proficiency of energy use in cogeneration mode can be up to 85 percent or sometimes higher.

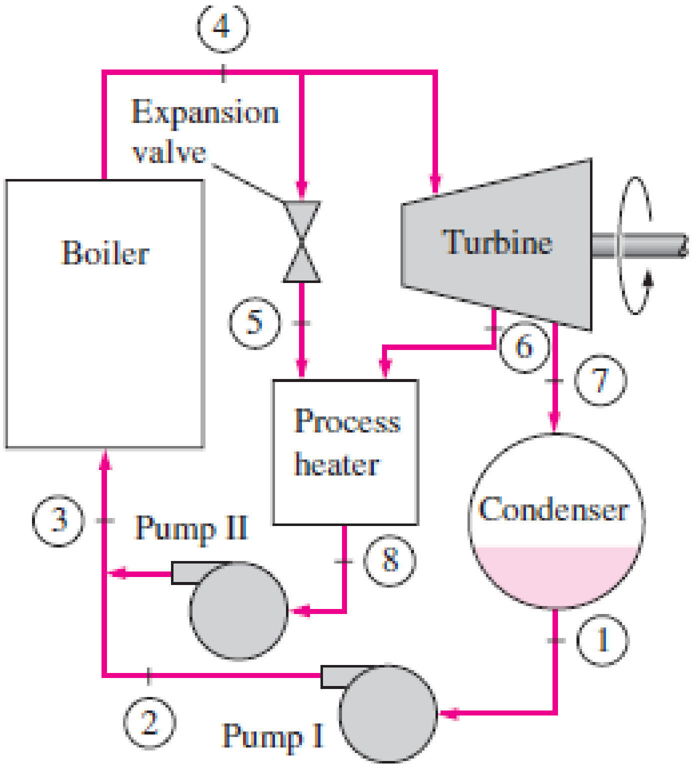


Figure 1.7: **Principle of Cogeneration[3]**

1.2.4 Combined Cycle

The journey for higher thermal efficiencies has brought innovative modifications to conventional power plants. A prominent change includes a Gas power cycle topping a Vapor power cycle, which is known as the combined Gas–Vapour cycle, or simply the combined cycle. The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam turbine (Rankine) cycle. The combined cycle has a higher thermal efficiency than both the cycles executed independently in the Fig 1.8 .

Gas-turbine cycles regularly work at impressively higher temperatures than steam cycles. The greatest fluid temperature at the turbine inlet is around 620°C for current steam power plants, however more than 1425°C for gas-turbine power plants. It is more than 1500°C at the burner exit of turbojet engines. The utilization of higher temperatures in gas turbines is made conceivable by improvements in cooling the turbine edges and covering the blades with high-temperature-resistant materials such as ceramics.

Because of the higher average temperature at which heat is supplied, gas-turbine cycles have a greater potential for higher thermal efficiencies.

• Typical Auxiliary Power Consumption of Combined Cycle = 2.5% to 3% of the Turbine Output.

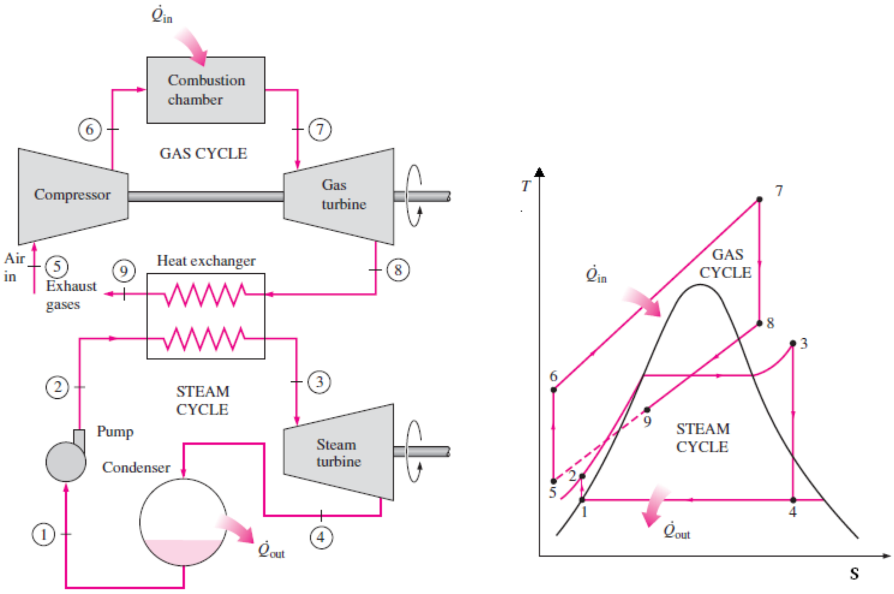


Figure 1.8: **Schematic with T-S diagram of Combined Cycle[3]**

1.2.5 Super-critical Cycle

Figure 1.9 demonstrates T-S chart for a super critical and subcritical steam cycle. Steam is generated in a "once-through" boiler at a pressure over the critical point of 221.2 bar, 371 °C Temperature. If the plant incorporates reheat and several stages of feed water heating, there is about a 2% gain in thermal efficiency compared with corresponding subcritical cycle.

Type	Pressure range
Subcritical	below 221.2 bar
Supercritical	221.2 to 250 bar
Ultra supercritical	Above 251 bar

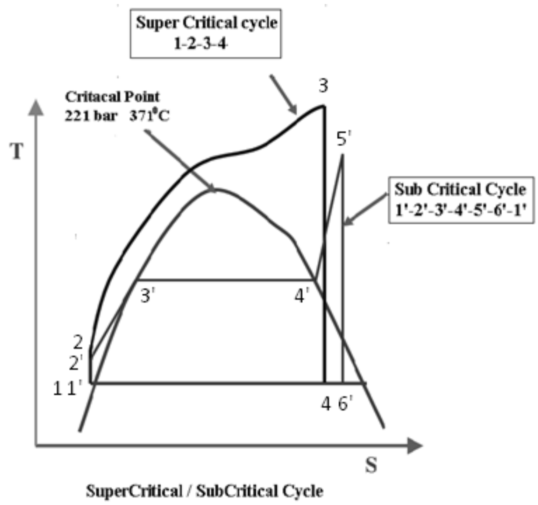


Figure 1.9: **T-S diagram of Subcritical and Super-critical Cycle**[1]

In the super critical cycle fuel, feed water and auxiliary power consumption are less, which increase plant efficiency. The size of condenser, cooling tower and coal handling system size also reduced in super critical cycle.

1.3 Objectives of the present study

The main objectives of the present work are as following :

1. To reduce / optimize the effective heat transfer area in a surface condenser a given application.
2. To reduce / optimize the tube side pressure drop in a surface condenser for a given application.
3. To study / analyse the effect of addition of a Desuperheating Zone in a LP Feedwater Heater.

Chapter 2

Literature Review

The literature review related to power plant heat exchangers such as surface condenser and feedwater heater is summarized in this chapter.

2.1 Power Plant Heat Exchangers

A heat exchanger is a device that is utilized to exchange thermal energy between two or more fluids, between a solid surface and a fluid or between solid particulates and a fluid at various temperatures in thermal contact. In heat exchangers, there are normally no externally heat or work associations. Common applications include heating or cooling of a fluid stream of concerns and evaporation or condensation of single or multi part fluid streams[2].

Heat exchangers are used to promote thermal energy flows at intermediate stages in process engineering, or as a final heat release to the environment, ambient air in most cases, which renders non-contact devices such as Shell and Tube heat exchanger and Plate heat exchanger are rather inefficient and a recourse is to be made of contact heat exchangers as the wet cooling towers treated aside. A special case is that of marine engineering, where seawater is available in the environment, greatly alleviating the thermal problem for heat-exchangers, but at a cost in materials compatibility (cupro-nickel or titanium must be used instead of copper or aluminium), since seawater is very corrosive with plenty of microorganisms[3]. In order to mitigate the effects of seawater on heat exchangers, and to minimise hull-pass-throughs, only one central heat exchanger is cooled by seawater (a PHE usually), and all other required heat exchangers use clean fresh-water as an intermediate fluid loop to finally discharge the energy at the seawater exchanger (centralised cooling system); different fluid loop layouts can be used, normally grouping several thermal loads by proximity of location and by temperature level.

In different applications, the target may be to recuperate or to reject heat or disinfect pasteurize, fractionate, distill, concentrate, crystallize or control a process fluids. In few heat exchangers, the fluids exchanging heat are in direct contact. In most heat exchangers, the heat exchange between fluids happen through a wall or into or out of a wall in a transient way. In many exchangers, the fluids are isolated by a heat transfer surface and ideally they don't mix/leak.

In contrast, exchangers in which there is intermittent heat exchanger between hot and cold fluid via thermal energy storage and release through the exchanger surface or matrix are referred to as indirect transfer type or simply regenerators.

Such exchangers usually have a fluid leakage from one fluid to stream to the other, due to presure differences and matrix rotation / valve switching. Common example of heat exchangers are shell and tube heat exchangers, automobile radiators, condenser, evaporators, air preheaters, cooling Tower, etc.

2.2 Heat Exchanger : Types

The various types of heat exchangers are classified as :

• Shell-and-tube heat exchanger, where one stream goes along a group of tubes and the other inside an external shell, parallel to the tubes, or in cross-stream.

• Plate heat exchanger, where creased plates are held in contact and the two liquids stream flow independently along neighboring channels in the design.

• Open-flow heat exchanger, where one of the streams is not confined within the equipment. They start from air-cooled tube-banks and are mainly utilized for conclusive heat discharge from a fluid to ambient air, as in the car radiator, additionally utilized as a part of vaporizers and condensers in air-conditioning and refrigeration applications, and in specifically fired home water heaters. At the point when gases flow along both sides, the overall heat transfer coefficient is extremely poor, and the best arrangement is to make utilization of heat pipes as intermediate heat exchange devices between the gas stream; generally, finned isolating surfaces, or, better, direct contact through strong recuperators are utilized[2].

• Contact heat exchanger, where two fluids go into direct contact. Besides, the contact can be continuous, i.e. at the point when the two liquids combine and after that differ by gravity constraints, in a cooling tower, or the contact can be then again with a third medium, normally solid as in the regenerative heat exchangers. At the point when the heat exchange process between the hot and cold fluids is deferred altogether, the term 'thermal energy storage' is utilized rather than RGE. There is constantly some contamination by entrainment of one liquid by the other, although commonly it is irrelevant or even planned. The various types of heat exchangers are shown in fig 2.1 .

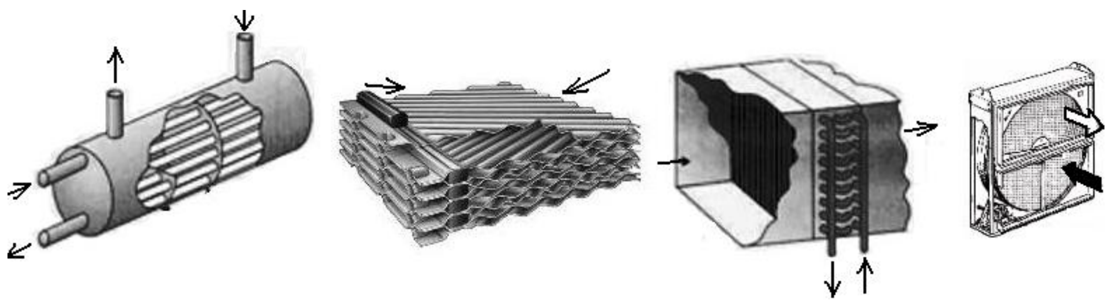


Figure 2.1: **Types of heat exchanges: a) shell-and-tube, b) plates, c) open-flow, d) rotating-wheel.[2]**

2.3 Surface Condenser

The most proficient and popular condensing systems are the water cooled surface condenser and used in areas where a lot of cooling water is available. A surface condenser is a shell and tube type indirect contact heat exchanger in which the steam flows inside the shell and cooling water flows inside the tubes. The hot fluid must be steam and cold fluid must be water. The exhaust steam from the turbine flows on the shell side of the condenser, while the plant's circulating water flows in the tube side. The source of circulating water can either be a closed loop or once through. The condensed steam from the turbine called condensate is gathered at base of condenser called hot well. The condensate is then pumped back to the steam generator to repeat the cycle[4].

The primary heat transfer mechanism in a surface condenser is the condensing of saturated steam on outside of the tubes and heating of the circulating water inside the tubes. Subsequently for a given circulating water flow rate, the water inlet temperature to the condenser decides the working pressure of the condenser. As this temperature decreases, the condenser pressure decreases. As described above, this decrease in pressure will increase the plant output and efficiency.

Because of the way that a surface condenser works under vaccum, non-condensable gases will move towards the condenser. The non-condensable gases comprise of generally air that has leaked into the cycle from segments that are working below atmospheric pressure. These gases can also result from the decomposition of water into oxygen and hydrogen by thermal and chemical reaction. These gases must be vented from the condenser otherwise the gases will increase the operating pressure of the condenser. Since the total pressure of the condenser will be the sum of partial pressures of the steam and gases, as more gas is leaked into the system, the condenser pressure will rise. This rise in pressure will decrease the turbine output and efficiency. The gases will cover the external surface of the tubes. This will decrease the heat transfer to the steam to the circulating water. Again the pressure in the condenser will increase the corrosiveness of the condensate in the condenser as the oxygen content increases[4].

Figure 2.2: **Steam surface condenser**

Oxygen causes corrosion, mostly in the steam generator. Thus these gases must be evacuated to extend the life of cycle parts. The two main devices that are utilized to vent the non-condensable gases are the Steam Jet Air Ejector and the Liquid Ring Vaccum Pump. The Steam Jet Air Ejectors utilize high pressure motive steam to evacuate the non-condensable gases from the condenser. The Liquid Ring Vaccum Pumps utilize a liquid compressant to compress the evacuated non-condensable gases and then release them to the atmosphere[5].

The air removal equipment must operate in two modes: hogging and holding. Before admitting the exhaust steam to a condenser, all the non-condensables must be vented from the condenser.

Now in the hogging procedure, a huge volume of air is quickly removed from the condenser by reducing the condenser pressure from atmospheric to a predetermined level.

Once the required pressure is accomplished, the air removal system will work in holding mode to evacuate all the non-condensable gases.

The 21 century is forming into a perfect energy storm. This is evidence from rising energy prices, diminishing energy availability and growing environmental concern. All these factors are quickly changing the global energy panorama. Energy and water are the key to modern life and they provide the basic necessities for sustained economic development. Industrialized nations have become increasingly dependent on fossil fuels. Securing sustainable and future energy supplies will be the greatest challenge faced by all nations in this century.

It is important that fossil fuel plants reduce their negative environmental impact by operating more efficiently. However with the increasing demand for one of the world essential commodity, the need for the optimization and increasing the efficiency of power plant performance arises. Generally, the performance of thermal power plant is evaluated through energetic performance criteria based on 1^{st} law of thermodynamics including electric power and thermal efficiency. In recent decades, the energetic performance based on second law of thermodynamics found a useful method for design , evaluation, optimization and improvement in thermal power plant[30].

2.4 Literature review for surface condenser

Prachi Bhatnagar et al[9]. had considered the empirical correlations and gave a design procedure of a surface condenser with the consideration of all critical parameters which directly or indirectly affects the condenser performance. The main focus of that work was on pressure drop inside the condenser by changing various parameters like velocity of water, overall design coefficient etc.

Ajeet Singh Sikarwar et al[10]. had studied about the factors or parameters which reduces the efficiency of condenser by using analysis of “Amarkantak Thermal power station” and worked on three causes which affecting the performance of condenser are deviation due to inlet temperature of cold water, deviation due to cold water flow, deviation due to air ingress / dirty tube. In this study, the efficiency of a power plant reduces to 0.4% by deviations in the condenser.

Milan Sanathara et al[11]. had worked on parametric analysis of surface condenser. Steam power plant mainly depends on the cold end operating conditions. The cooling system is considered as once through based on the thermodynamic model. The authors conclude that as the cooling water flow is lower, it leads to increase condenser pressure than design value and as a result poor heat transfer will occur whereas the higher water flow rate leads the condensate under cooling which is not useful because it decreases the cycle efficiency.

Nirmalkumar P. Bhatt et al[12]. had used genetic algorithm for the cost optimization of steam surface condenser. The mathematical model for the theoretical design and the cost calculation is given. For optimization, the authors had taken three different case studies and reduced the heat transfer area same heat duty and results had been compared for all three approaches.

Gokhan Ozdamar et al[13]. had worked on optimization of condenser in thermal power plant. The effect of condensation on condenser area and simultaneously on the cost is investigated. The change in the rate of condenser area relative to the condenser pressure drop is investigated. It is concluded that as the condenser pressure decreases, the condenser area increases and the cooling water pump power also increases and simultaneously cost increases.

Amir Vosough et al[14]. had worked on improvement of power plant efficiency with condenser pressure. The authors had done parametric study and exergy analysis of condenser. They found that decreasing the cycle condenser pressure and temperature will result to higher power output for the same mass flow rate of steam and fuel input to boiler, resulting in higher work output of turbine. The second parameter is the turbine outlet enthalpy which is a function of condenser pressure. Decreasing the turbine output enthalpy causes the turbine output work to increase and for increasing the turbine work, condenser pressure should be reduced.

Vikram Haldkar et al[15]. highlighted that the causes which effecting the performance of condenser are deviation due to cooling water inlet temperature, deviation due to water flow rate and deviation due to condenser pressure in energy efficiency of plant are included. Eventually the authors found the total efficiency of power plant will reduce 2.7% by all these deviations.

N. Naga Varun et al[30]. had worked on exergy analysis of a 210 MW plant of Vijayawada Thermal power station and found that the maximum exergy destruction was observed in Low pressure turbine which shown that there is scope to develop the efficiency.

Table 2.1: **Summary of Literature Review for Condenser**

Sr no.	Title	Authors	Remarks
1.	Surface Condenser-A Review	Prachi Bhatnagar, V. N bartaria	• Design procedure for condenser considering all critical parameters.
2.	Performance Analysis of Surface Condenser under Various Operating Parameters	Ajeet Singh Sikarwar, Devendra Dandotiya, Surendra kumar Agarwal	• Deviation due to Temperature of cold water, deviation due to cold water flow and deviation due to dirty tubes.
3.	Parametric Analysis of Surface Condenser for 120 MW Thermal Power Plant	Milan V. Sanathara, Ritesh P. Oza, Rakesh S. Gupta	• As the cooling water flow is lower, it leads to increase condenser pressure then the design value which results in poor heat transfer.
4.	Steam Surface Condenser Design based on Cost Optimization using Genetic Algorithm	Nirmalkumar P. Bhatt, A. M. Lakdawala, V. J. Lakhera	• The mathematical model for theoretical design and cost calculation procedure given. Three different case studies had taken and improved heat transfer for same heat duty.
5.	Optimization of Condenser in Thermal Power Plant	Gokhan Ozdamar, Recap Ozturk	• The effect of condensation on the condenser area and on the cost. The change in the rate of condenser area relative to condenser pressure drop.
6.	Improvement in Power Plant efficiency with condenser pressure	Amir Vosough, Alireza Falatah, Sadeg Vosough, Hasn Nasr Esfehani, Azam Behjat, Roya Naseri Rad	• Decrease in cycle condenser pressure will result to higher power output for same mass flow rate of steam.
7.	Parametric Analysis of Surface Condenser for Thermal Power Plant	Vikram Haldkar, Abhay kumar Sharma, R. K. Ranjan, V. K. Bajpai	• The total efficiency of power plant will reduce 2.7% due to the deviations in cooling water flow rate, Cooling water inlet temperature.
8.	Exergy Analysis of 210 MW Vijaywada Thermal Power Station	N. Naga Varun, G. Satyanarayan	• The maximum exergy destructin was observed in low pressure turbine. The exergy destruction of condenser is 19 MW.

2.5 Feedwater Heaters

A Feedwater Heater is a power plant component used to preheat the water delivered to a steam generating boiler. A practical regeneration process in the steam power plant is accomplished by extracting or bleeding steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine is used to heat the feed water instead. The device where the feed water is heated by regeneration is called the Regenerator or the Feedwater Heater. A Feedwater Heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two streams or without mixing them[6].

Open Feedwater Heater :

An Open Feedwater Heater is basically a mixing chamber where the steam extracted from the turbine mixes with the feedwater exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure. The advantages of open heater are simplicity, lower cost and high heat transfer capacity. The disadvantage is the necessity of a pump at each heater to handle large feedwater stream.

Closed Feedwater Heater :

In Closed Feedwater Heater, the heat is transferred from the extracted steam to the feedwater without mixing taking place. The feedwater flows through the tubes in the heater and the extracted steam condenses on the outside of the tubes in the shell. The heat released from the condensation is transferred to the feedwater through the walls of the tubes. The condensate, sometimes called the heater-drip passes through a trap into the next lower pressure heater. To some extent, this reduces the steam required by that heater. The trap passes only liquid and no vapours. The drip from the lowest pressure heater could similarly be trapped to the condenser, but this would be throwing away energy to the condenser cooling water. To avoid this waste, the drip pump feed the drip directly into feedwater stream.

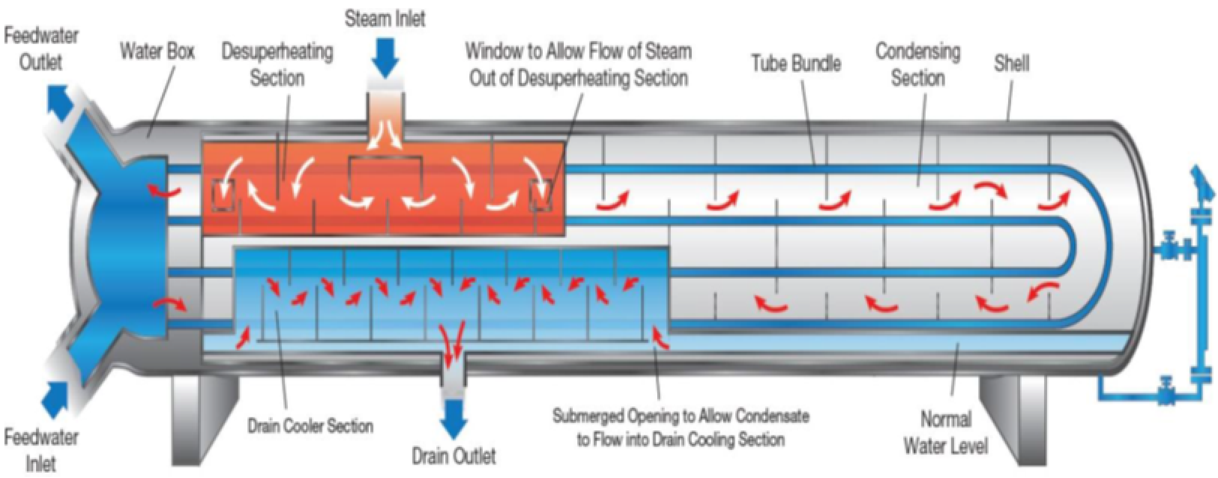


Figure 2.3: **Closed Feedwater Heater**

A Closed Feedwater Heater system requires only a single pump for the main feedwater stream. Closed Feedwater Heaters are costly and may not deliver as high feedwater temperature as with open feed water heaters. In most steam power plants, closed heaters are preferred but at least one open heater is used for the purpose of feedwater deaeration. The open heater in such a system is called Deaerator.

A Feedwater Heater is a heat exchanger designed to preheat boiler feedwater by means of condensing steam extracted from steam turbine. The heaters discussed here are known as closed since the tube side fluid remains enclosed by the tubes and channel and does not mix with the condensate as is the case with open feedwater heater. They are unfired since the heat transfer within the vessel does not occur by means of combustion but by convection and condensation[6].

The steam extraction process in closed feed water is referred to as uncontrolled extraction. The flow rate of steam into a feedwater heater is not limited by the amount of available steam. The shell side operating pressure in a feedwater heater is determined by the pressure of the steam supplied to it and not by the amount of heat transfer surface.

2.6 Literature review for Feedwater Heater

Mario Alvarez-Fernandez et al[20]. had introduced a model for the thermal analysis of feedwater heaters in which wet steam is extracted from the steam turbine. The model has relevant applications in nuclear power plants, where the fluid flowing through the steam turbine is wet steam.

Sergio Espatolero et al[21]. had worked on feedwater heater arrangement and flue gas heat recovery system analysis and optimization by thermodynamic approach and economic approach. A case application study has shown that with the new LP heater addition, two drain pumps and the indirect flue gas heat recovery system with a double stage integration with the cycle, raise overall power plant efficiency upto 0.7 points and cost analysis has confirmed that the utilization of these systems is also a profit making alternative.

E. Devandiran et al[27]. had studied the effect of the feedwater heaters on the performance of coal-fired power plant through the analysis of using HMBD. An analysis shows that overall power plant efficiency has been increased by 2.4 % using feedwater heater. The turbine heat rate and coal consumption have been considerably decreased. This implies a direct reduction CO_2 emission. The final result shows an overall better performance of power plant by using feedwater heaters.

Suhas D. Ambulgekar et al[29]. highlighted that the maximum amount of heat duty is handled in the condensing zone of feedwater heater. Shell side heat transfer coefficient has a large influence on overall heat transfer coefficient and hence area of feedwater heater. Fouling factor affects in calculation of surface area to a large extent. Condensing zone accounts for maximum amount of area of feedwater heater than other zones. Pressure drop on shell side is maximum in subcooling zone as compared to other two zones. While pressure drop on the tube side of feedwater heater is negligible.

Table 2.2: **Summary of Literature Review for Feedwater Heater**

Sr no.	Title	Authors	Remarks
1.	Thermal Analysis of Closed Feedwater heaters in Nuclear Power Plants	Mario Alvarez-Fernandez, Luis del portillo-Valdes, Christina Alonso-Tristan	• Introduced model for the thermal analysis of feedwater heater in which wet steam is extracted from the steam turbine.
2.	Efficiency Improvement Strategies for the Feedwater heater network designing in Super critical Coal-fired Power Plants	Sergio Espatolero, Luis M. Romeo, Cristobal Cortes	• By the addition of new LP heater, two drain pumps and the indirect flue gas heat recovery system, the overall power plant efficiency increases upto 0.7 % and cost analysis also confirmed that the utilization of these system is also a profit making alternative.
3.	Influence of Feedwater heaters on the performance of Coal-fired Power Plants	E. Devandiran, V. S.Shaisundaram, P. Sabari Ganesh, S. Vivek	• The overall power plant efficiency has been increased by 2.4 % using feedwater heaters.
4.	Design and Analysis of High Pressure Feedwater Heater	Suhas Ambulgekar, B.S gawali, N. K. Sane, Vinaykumart Kabra, Ajit Gavali	• Maximum amount of heat duty is handled in the condensing zone of feedwater heater. Condensing zone accounts maximum amount of area of feedwater heater than other zones.

Chapter 3

Design of Power Plant Heat Exchanger

The general procedure for the design of power plant heat exchanger is described in this chapter.

3.1 Condenser Design Methodology

The heat load to be used in designing a condenser is the energy that must be removed from the exhaust steam to cause it to condense[1].

The optimum condenser operating pressure must be determined in conjunction with the steam turbine. Limitations imposed by the circulating water parameters of flow, inlet temperature and temperature rise initiate lowest possible condenser pressure.

The lowest possible pressure may not be the most economical from the view point of the equipment cost, operating cost and efficiency. An economic analysis is required to determine the optimum condenser operating design pressure.

It is recommended that the operating pressure be limited by not allowing the design Terminal Temperature Difference to be less than 5 °F (2.8 °C)[5].

Generally the tube arrangement inside the condenser shell is triangular arrangement. Hence during the working condition the condenser dome is fully occupied with steam, the surface of tube is fully in contact with the steam from all direction, the mass quality of steam is considered nearly equal to 1 and the condenser is working under full vaccum condition is assumed here.

3.1.1 Heat load and cooling water flow calculations

Heat load calculation formula is given as following :

$$Q = \sum m \times C_p \times TR = \sum m \times h \tag{3.1}$$

where
m = mass flow rate of steam from various equipments
h = Enthalpy of steam
TR = Temperature rise = $T_2 - T_1$

The steam inlet from various devices is listed as following :

1. Steam from main turbine exhaust
2. Steam from BFP drive turbine
3. Feed water heater drains including steam seal leak off
4. Gland steam condenser

The cooling water outlet temperature calculation formula given as below:

$$T_2 = T_s - TTD \tag{3.2}$$

Ts = Temperature at condenser operating pressure
TTD = Terminal Temperature difference

The cooling water volumetric flow rate formula is as given below:

$$M = \frac{Q \times V}{C_p \times TR} \tag{3.3}$$

V = Specific volume of cooling water
M = Total cooling water volumetric flow

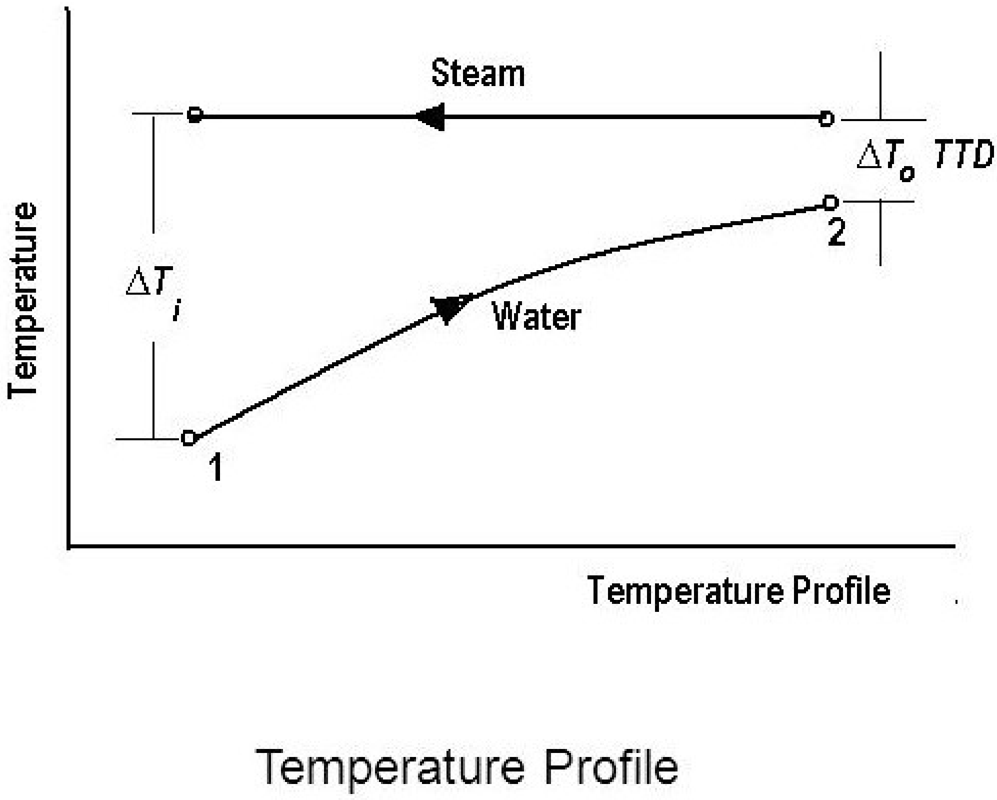


Figure 3.1: **Temperature profile for Condenser**[4]

3.1.2 Heat transfer area calculation

The overall heat transfer coefficient is calculated by using the following formula:[5]

$$U = U_1 \times F_m \times F_c \times F_w \tag{3.4}$$

U_1 = Uncorrected overall heat transfer coefficient which is taken from HEI standard on the basis of the value of cooling water velocity inside the tube and tube diameter.
F_m = Material correction factor which is taken from the HEI standard on the basis of selected tube material and thickness of tube.
F_c = Cleanliness factor generally consider between 0.75-0.9 according to condition of cooling water flow.
F_w = inlet water temperature correction factor which is taken from HEI standard on the basis of cooling water inlet temperature.

The LMTD is calculated as the following :

$$LMTD = \frac{T_2 - T_1}{ln\frac{(T_s - T_1)}{(T_s - T_2)}} \tag{3.5}$$

The effective heat transfer area is calculated as the following :

$$A = \frac{Q}{U \times LMTD} \tag{3.6}$$

3.1.3 Number of tubes per pass and Length of tube calculations

For finding out number of tubes per pass, there are many formulas available. The HEI standard's formula is calculated as following :[5]

$$N_t = \frac{Total\ cooling\ water\ volumetricflow}{Velocity\ of\ water\ in\ tube \times volumetric\ flow\ per\ m/s\ velocity\ per\ tube} \tag{3.7}$$

The volumetric flow per m/s per tube value is given in HEI standard on the basis of tube outer diameter and tube thickness.

For calculating the number of tubes, the following equation may also be used :

$$N_t = \frac{Heat\ trasnfer\ area}{\Pi \times Outer\ tube\ diameter \times Length\ of\ tube} \tag{3.8}$$

The condenser is mainly divided into two different zones.

At the top of the condenser where the steam inlet in the condenser is provided and steam impacts on condenser is known as impingement zone. In the impingement zone the vaccum generators or air removal devices are placed. Some percentage of the total number of tubes are placed in the impingement zone.

The remaining part of the condenser is known as the condensing zone. The steam condensation process happens in the condensing zone only and it is around 80% of the total area.

The length of tube is calculated by the above equation only after knowing the number of tubes. Sometimes due to space constraint the length of tube is already defined from the site. So according to that we have to calculate the number of tubes.

3.1.4 Total hydraulic loss calculation

The circulating water pressure loss through the condenser is calculated by using the following equation.

Total hydraulic loss:[5]

$$R_{tt} = (R_1 \times R_2 \times R_t \times L) + R_{iw} + R_{ow} + R_{te} \tag{3.9}$$

$$R_2 \times R_t = \frac{0.00642 \times V_w^{1.75}}{D_t^{1.25}} \tag{3.10}$$

R1 = Temperature factor
Riw = Water box inlet loss
R2 = Tube O.D. and Gauge factor
Row = Water box outlet loss
Rt = Tube friction loss
Rte = Tube loss (Inlet + Outlet)

This is one method to design condenser which is followed by the industry and which is also given in HEI standard for steam surface condensers. There are also other methods which are known as Kern method and Bell Delaware method for designing a heat exchanger or steam surface condenser.

Fig. 3.2 and Fig. 3.3 cover the head losses to be expected in the water boxes, tube entrances, exits of single pass condenser and two pass condensers respectively. The inlet and outlet water box losses should be determined from the curve in fig. 3.3 using the actual nozzle water velocity in each case.[5]

The tube inlet and outlet losses are combined in one curve and value of those losses should be taken directly from the curve using the actual water velocity in the tubes.

For two pass the tube losses become double and similar procedure should be used for three and four passes.

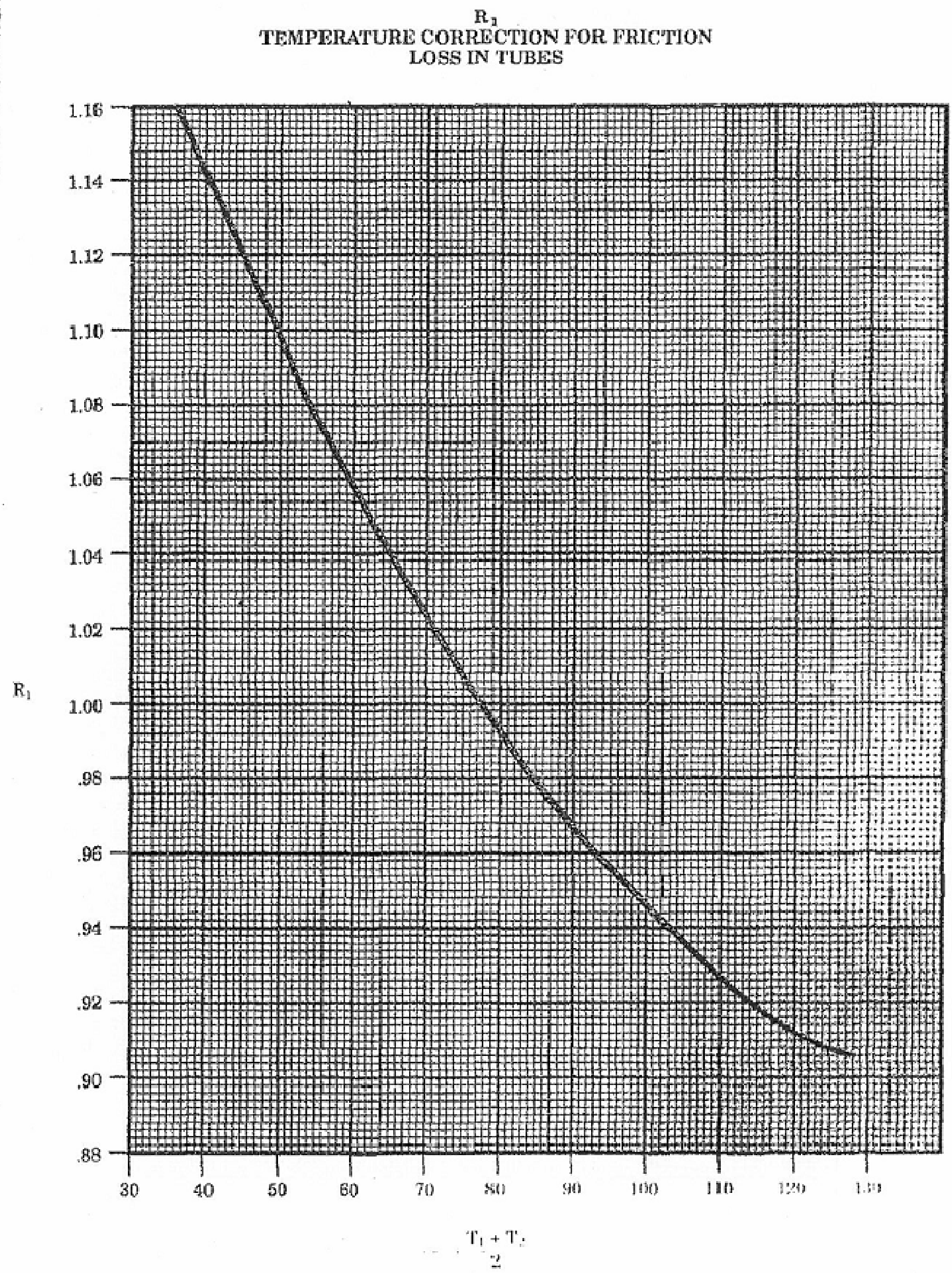


Figure 3.2: **Temperature correction for friction losses in tubes[5]**

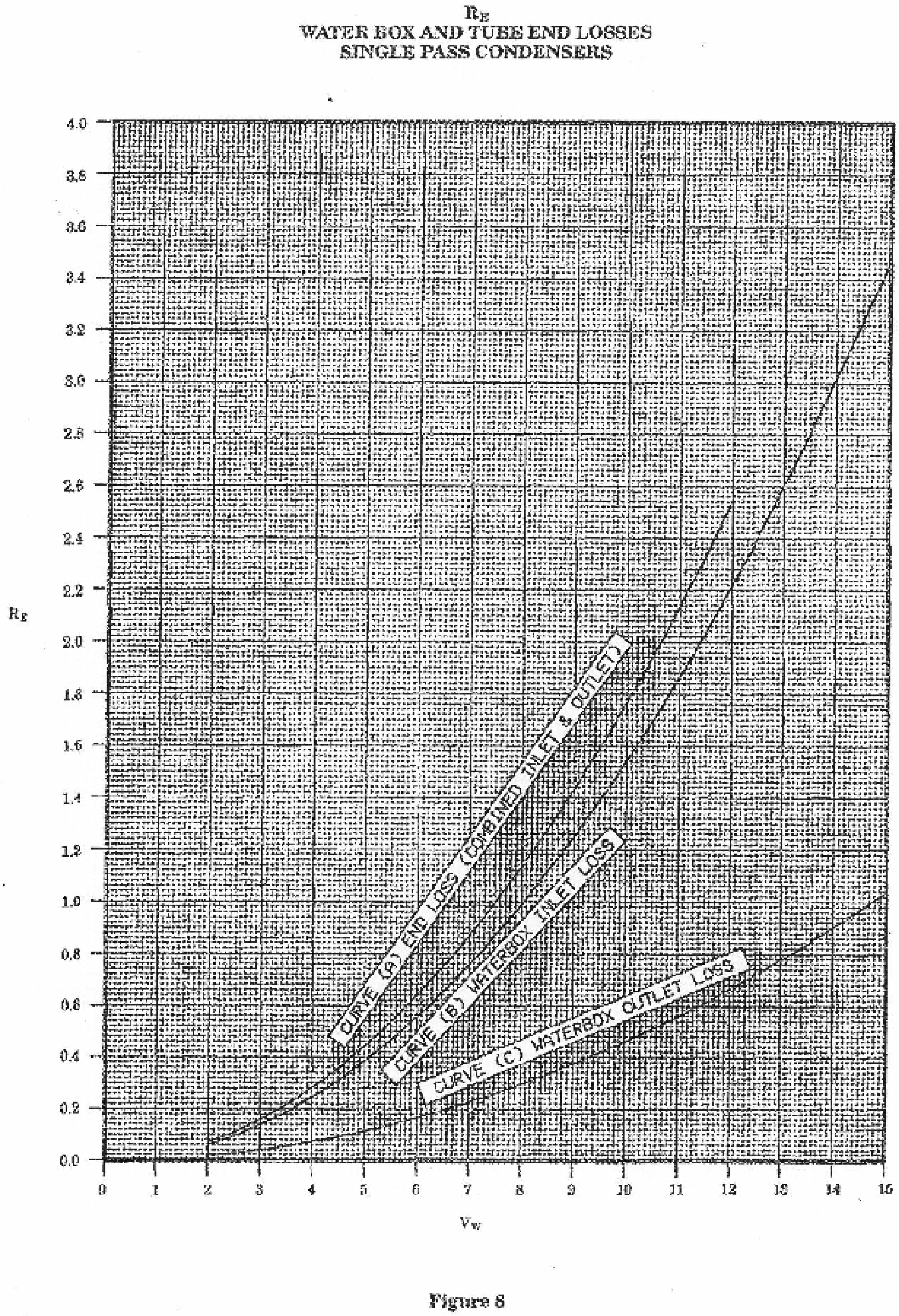


Figure 3.3: **Water box and tube end losses for single pass[5]**

3.2 Closed Feedwater Heater Design Procedure

It is recognized that the performance of a feedwater heater cannot be exactly predicted under a number of possible operating conditions; therefore, the heater should be designed for the one specific condition termed as the " design point ".[6]

The heater design performance is stated as the capability to heat a given flow of feedwater in terms of TTD and DCA, if applicable with the following parameters specified :

(a) Feedwater inlet and outlet temperatures
(b) Drain outlet temperature
(c) Steam pressure and enthalpy
(d) Feedwater pressure loss
(e) Shell side pressure loss

According to HEI Standard, it is recommended that heaters without desuperheating zones should not be designed for a Terminal Temperature Difference of less than 2 °F (1.12 °C)[6].

The performance of integral subcooling zone is dependent upon many factors such as: heater orientation, feedwater temperature rise, drain cooling range, quantity of drains and reheating of subcooled condensate. Since experience has determined that the closest approach that can be assured is 10 °F (5.6 °C), it is recommended that heaters not be designed for less than that temperature approach.

The overall pressure loss should not exceed 30 percent of the differential pressure between heater stages.

The pressure loss within any one zone should not exceed 5 psi (34.4738 kPa).

3.2.1 Desuperheating Zone

In the desuperheating section, the superheated steam is cooled in a contact with dry tubes of the U-tube bundle to the saturation temperature.

(A) Thermal analysis for tube side [2]

1. Number of Tubes :

$$m_t = \rho_t \times u_t \times A_c \times N_t \tag{3.11}$$

2. Reynolds number :

$$Re = \frac{\rho_t \times u_t \times d_i}{\mu_t} \tag{3.12}$$

,

$$Pr = \frac{\mu_t \times Cp_t}{k_t} \tag{3.13}$$

3. Nusselt number :

$$Nu_t = \frac{(f/2) \times Re \times Pr}{1.07 + [12.7 \times (f/2)^{1/2} \times (Pr^{2/3} - 1)]} \tag{3.14}$$

Where f is the friction factor which can be calculated from

$$f = (1.58 \ \ lnRe - 3.28)^{-2} \tag{3.15}$$

4. Tube side heat transfer coefficient :

$$h_t = Nu \times \frac{k_t}{di} \tag{3.16}$$

(B) Thermal Analysis for Shell Side [2]

1. Total flow area through tubes :

$$A_t = \frac{\pi \times di^2 \times N_t}{4} \tag{3.17}$$

2. Shell inside diameter :

$$D_s = 0.637 \times \sqrt{\frac{CL}{CTP}} \times (\pi \times do^2 \times PR^2 \times N_t)^{1/2} \tag{3.18}$$

CL = 1.0 For Square Pitch

CL = 0.87 For Trinagular Pitch

PR = Pt / do

CTP = 0.93 for one-tube pass

CTP =0.90 for two-tube pass

CTP =0.85 for Three-tube pass

3. Shell equivalent diameter :

$$De = \frac{4 \times \left(\frac{\sqrt{3}}{4} \times P_t^2 - \frac{\pi \times do^2}{8} \right)}{\frac{\pi \times do}{2}} \tag{3.19}$$

for Triangular Pitch

4. Shell-side Reynolds number :

$$Re_s = \left(\frac{m_h}{A_s} \right) \times \frac{De}{\mu_h} \tag{3.20}$$

$$A_s = \frac{D_s \times C \times B}{P_t} \tag{3.21}$$

C = Tube Clearance

B = Baffle Spacing

5. Shell-side heat transfer coefficient :

$$h_o = \frac{0.36 \times k_h}{De} \times Re_s^{0.55} \times Pr_h^{1/3} \tag{3.22}$$

(C) Overall Heat Transfer Coefficient

$$U_f = \left(\frac{1}{h_o} + \frac{do}{di \times h_i} + R_{ft} + \left(\frac{do}{2} \times \frac{ln\left(\frac{do}{di}\right)}{k_t} \right) \right)^{-1} \tag{3.23}$$

3.2.2 Condensing Zone

In the condensing zone, the steam condenses in contact with tubes where it changes its phase from saturated vapor to saturated liquid and releases the latent heat of vaporization.[29]

The heat transfer in the condensing zone is the latent heat exchange,

$$Q = m_s \times h_{fg} = m_t \times Cp_t \times (T_{co} - T_{ci}) \tag{3.24}$$

The driving force for condensation is the difference between the temperature of cold wall surface and the bulk temperature of the saturated vapor.

$$\triangle T_{driving} = T_{sat} - T_{wall} \tag{3.25}$$

The viscosity and other properties used in the condensing correlations are evaluated at the film temperature, a weighted mean of the cold surface temperature and the vapor saturation temperature,

$$T_{film} = T_{sat} - \frac{3}{4} \times (T_{sat} - T_{wall}) \tag{3.26}$$

where

$$T_{wall} = \frac{T_{sat} + \frac{T_{co}+T_{ci}}{2}}{2} \tag{3.27}$$

The heat transfer coefficient for condensation is given as,

$$h_{cond} = 0.725 \times \left\{ \frac{k_f^3 \times \rho_f \times (\rho_f - \rho_v) \times g \times h_{fg}}{\mu \times (T_{sat} - T_{wall}) \times do} \right\}^{1/4} \tag{3.28}$$

3.2.3 Subcooling Zone

In the subcooling zone, the tube side and shell side fluids are liquids i.e. single phase and the heat exchange occurs in form of sensible heat transfer. The values of heat transferred, LMTD, overall heat transfer coefficient, area of zone and length of tube can be calculated using same equations as that of the desuperheating zone.

Chapter 4

Parametric Analysis of Heat Exchanger

4.1 Parametric Analysis of Condenser

There are many parameters which affect the performance of the condenser. The pressure drop is one of the key and important parameters in the design of condenser as well as analysis of the condenser performance. The thermal efficiency of a Power Plant can be increased by increasing the condenser heat transfer rate. This can be achieved by reducing the pressure of turbine exhaust. Generally the condensation process occurs at a constant pressure as well as temperature. The shell side pressure drop has to be constant but actually the main reason of pressure drop occurrence is that the vaccum inside the shell cannot be maintained constant during the working of condenser. The tube side pressure drop generally occurs and rises due to improper heat transfer[1].

4.1.1 Key parameters affect in condenser performance

There are some key parameters which directly or indirectly affect the condenser pressure drop.

Velocity of water:

As the velocity of circulating water decreases the pressure drop reduces. As we know from the Bernoulli's equation that when liquid flows from higher pressure to lower pressure then the velocity of flowing fluid will affect on the heat transfer rate[5].

Overall design coefficient:

By lowering down the overall design coefficient, the pressure drop will also decrease. It indicates that when the overall design heat transfer coefficient will be lower,the number of tube required will be higher and the pressure drop decrease[5].

Outer diameter of tubes:

As the outer diameter of condenser tube is increased, the velocity of the cooling water flow decreases and consequently the pressure drop will also reduce.

Tube material:

The condenser tube material also affects the condenser performance. Some materials have higher thermal conductivity and so a higher heat transfer rate. But there is corrosion effect on material and life which affects the condenser performance. Even though if the material has a higher heat transfer coefficient but a lower corrosion resistance, it will not survive for a long time. So it cannot be preferred as a tube material. The material should have lower thermal expansion on temperature rise[12].

4.1.2 Calculation of condenser design for a 660MW Thermal power plant

The detailed calculation of condenser design is shown in Appendix A.
The input data for condenser design is shown in Table 4.1.

Table 4.1: **Input data[25]**

INPUT DATA	Unit	LP Shell@2.838
Condenser Pressure P_s	kg/cm^2 abs.	0.071
Condenser Temperature T_s	°C	38.92
Condenser Heat Duty, Q	kCal/hr	300297115.94
Circulating water flow rate, W_G	m^3/hr	67505.7364
Circulating water inlet temperature, T_1	°C	31.6
Terminal temperature difference	°C	2.838
Tube water Velocity, V_w	m/s	1.7474
Cleanliness factor, F_c	-	0.9
Tube Outer Diameter, D	mm	25.4
Tube Inner Diameter, D_1	mm	23.978
Tube pitch P_t	mm	31
Thickness of impingement zone @ 20 BWG	mm	0.889
Thickness of condensing zone @ 22 BWG	mm	0.711
Specific volume	m^3/kg	0.001005553
Tube material	-	SS TP 304
Circulating water type	-	Fresh Water
Circulating water density, r	kg/m^3	994.0
Circulating water Specific Heat, C_p	kCal/kg °C	0.99803
Number of Tube pass	-	1
external surface @22	m^2/m	0.0798
Length per surface per tube@22	m/m^2	12.523
water - m3/hr at 1 m/s velocity@22	$(m^3/hr)/(m/s)$	1.6219
mean temperature	°C	33.841
Temperature factor R_1	-	0.9622
R_2*R_t	-	0.14676
Average inner diameter	mm	23.95
Water box inlet loss	m	0.1899
Water box outlet loss	m	0.0601
Tube end losses	m	0.1771
Total hydraulic loss	m	2.414
Margin for fabrication and erection	%	0.5

The heat duty is calculated from the heat balance diagram of power plant and other required parameters are taken from HEI standard and the data provided by the company. The calculated parameters are shown in the result Table 4.2.

Table 4.2: **Calculated results**

Sr no.	**Result**	**Unit**	**LP Shell@2.838**
1	CW outlet Temp. T2	°C	36.082
2	LMTD	°C	4.7303
3	Overall Heat Transf. Coeff. U	kCal/hr m^2°C	2555.43
4	Heat Transfer surface Area A_s	m^2	24843
5	Temperature rise	°C	4.482
6	Final provided surface area	m^2	26756.11
7	Circulating water flow	m^3/hr	67505.7
8	Total number of tubes per pass	-	23765
9	Surface per tube	m^2	1.1233
10	Length of tube	m	14.12
11	Total hydraulic loss	mWc	2.419

The results are shown for TTD = 2.838 °C
The detailed calculations are as shown in Appendix A.

4.1.3 Comparison of results of HEI design method and KERN method

The comparison of results considering HEI design method and Kern method are as following :

Sr no.	**Particulars**	**HEI method**	**KERN method**
1	Overall heat transfer coefficient	2971.46	3046.61
2	LMTD	4.7303	4.7305
3	Heat transfer area	26756	26651
4	Number of tubes	23765	23749
5	Length of tubes	14.1	14.1
6	Shell diameter	5.22	5.22

4.1.4 TTD effect on condenser parameters

The effect of change of TTD on condenser parameters are tabulated in this section.
Case: 1 Constant heat duty = 300297115.94 kCal/hr

TTD	°C	2.7	2.838	3
LMTD	°C	4.5363	4.7303	4.8431
Length	m	15.12	14.07	13.24
Area	m^2	25905	24843	24264
Circulating water flow rate	m^3/hr	65489.3	67505.7	70037.2
Number of tubes	-	23108	23820	24713

Case: 2 Constant heat duty Q = 300297115.94 kCal/hr & Constant length L = 14.12 m

TTD	°C	2.7	2.838	3
LMTD	°C	4.5363	4.7303	4.8431
Length	m	14.12	14.12	14.12
Area	m^2	25905	24843	24264
Circulating water flow rate	m^3/hr	65489.3	67505.7	70037.2
Number of tubes	-	24838	23820	23177

The results show that for the constant heat duty if we increase the value of TTD, the length of condenser decreases, the effective heat transfer area also decreases but the number of tubes increases because % decrease in length is more than the % decrease in the heat transfer area. So according to number of tubes formula, the tube count will increase.

In the second case if we make the heat duty and the length of tube both constant then as TTD increases, the effective heat transfer area as well as the number of tubes also decrease.

There are many other parameters like tube material, tube diameter, tube thickness, tube velocity etc. which affect the condenser performance.

The effect of heat transfer area v/s tube material at different TTD values is shown in fig. 4.1 while the effect of tube length v/s tube material at different TTD values is shown in fig. 4.2.

Table 4.3: **Effect of TTD and tube material on effective heat transfer area & tube length**

Material	Heat transefer Area in m^2			Length of Tube in m		
	2.7	2.838	3	2.7	2.838	3
Cu-Fe 194	23169.00	22219.43	21702.05	12.56	11.68	11.01
Arsenical Cu	23281.70	22326.98	21807.09	12.62	11.74	11.05
Admiralty	23555.35	22589.40	22063.40	12.77	11.88	11.18
Al-brass	23624.77	22655.46	22128.40	12.80	11.91	11.21
Al-bronze	23835.49	22858.05	22325.80	12.92	12.02	11.31
Carbon steel	24465.93	23462.63	22916.31	13.26	12.34	11.61
Cu-Ni 90-10	24540.83	23534.46	22986.46	13.30	12.37	11.65
Cu-Ni 70-30	25422.85	24830.31	23812.61	13.78	12.82	12.07
SS(UNS S43035)	25971.84	24906.80	24326.92	14.07	13.09	12.33
Titanium grade 1&2	26284.17	25206.30	24619.38	14.24	13.25	12.48
SS(UNS S44660)	26992.16	25885.26	25282.52	14.63	13.61	12.81
SS(UNS S44735)	27144.48	26031.34	25425.20	14.71	13.69	12.88
SS TP 304	27900.25	26756.11	26133.10	15.12	14.07	13.24
SS TP 316&317	28161.61	27006.75	26377.90	15.26	14.20	13.37
SS(UNS N08367)	29222.37	28024.02	27371.48	15.84	14.73	13.87

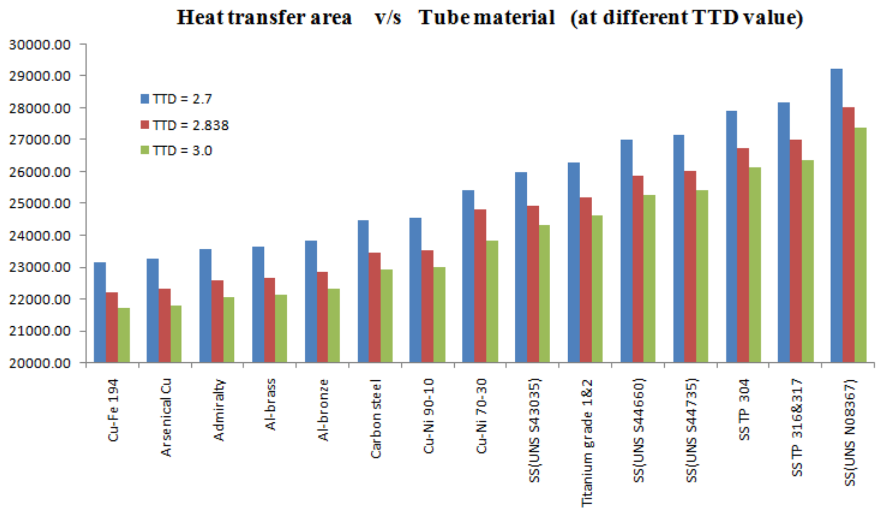


Figure 4.1: **Heat transfer area v/s Tube material (at different TTD values)**

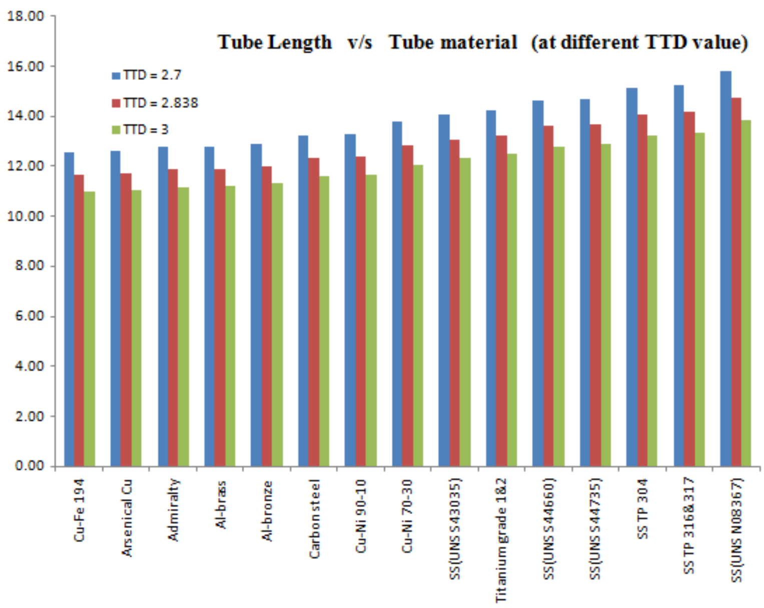


Figure 4.2: **Tube Length v/s Tube material (at different TTD values)**

4.1.5 Tube diameter effect on condenser parameters

Tube diameter is also one key parameter for the performance analysis of condenser. Because as tube diameter changes, velocity, pressure drop, number of tubes, length of tube etc. parameters will change. So it will change the condenser size and simultaneously it will effect on cost of condenser also.

Let us see some plots for tube hydraulic loss or we can say it as pressure drop, length and number of tubes with different tube outer diameters.

The effect of the change in the length of tubes with tube outer diameter is shown in fig. 4.3.

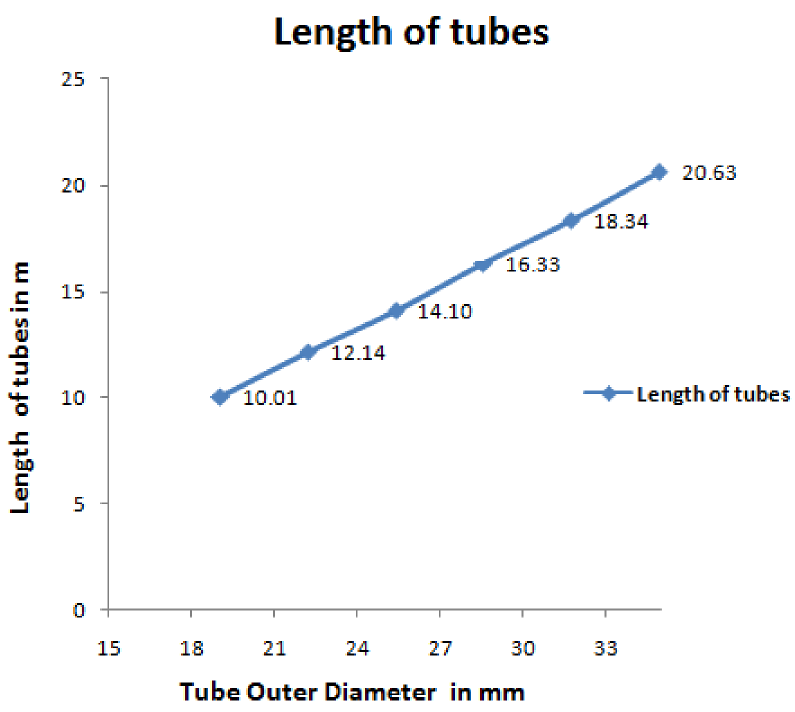


Figure 4.3: **Length of tube v/s Tube Outer Diameter**

The effect of Total hydraulic loss with tube outer diameter is shown in fig. 4.4.

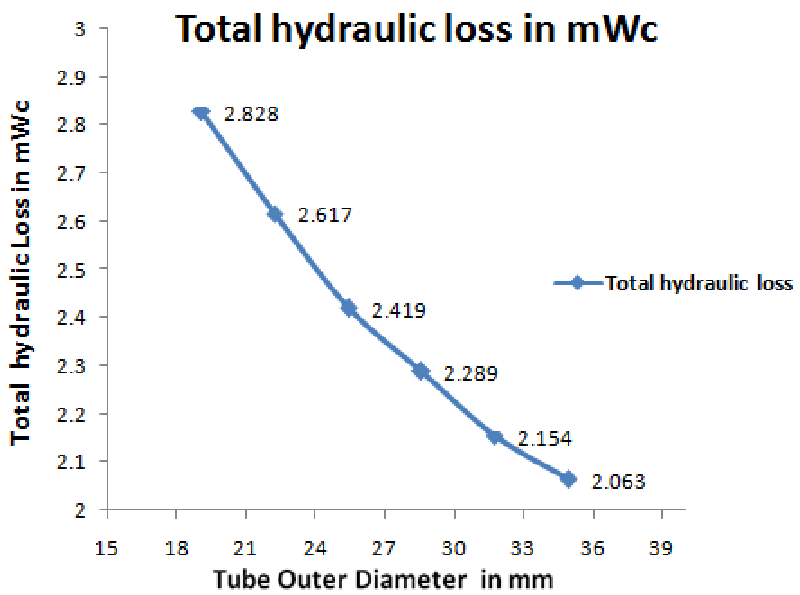


Figure 4.4: **Total Hydraulic Loss v/s Tube Outer Diameter**

The effect of the change in number of tubes with tube outer diameter is shown in fig 4.5.

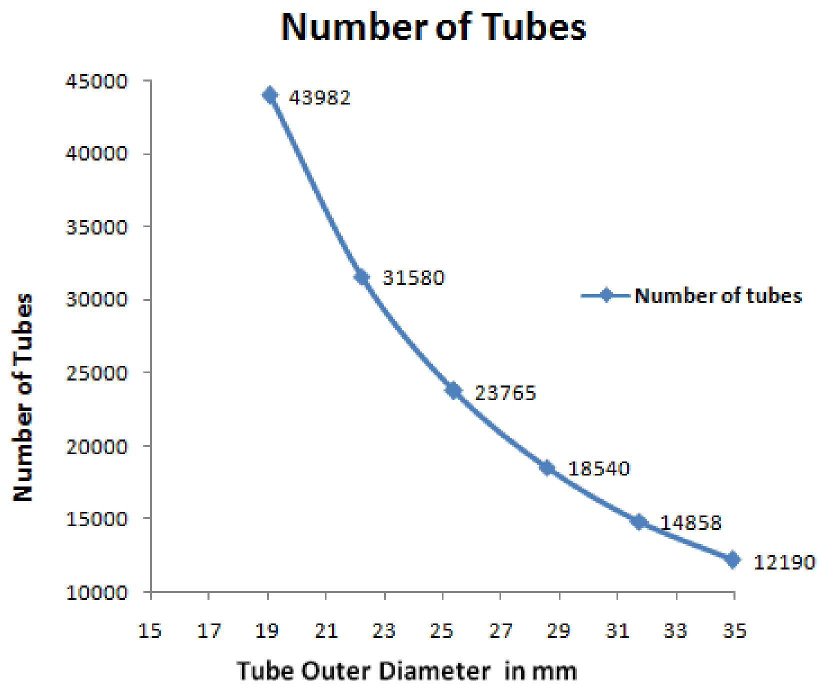


Figure 4.5: **Number of Tubes v/s Tube Outer Diameter**

Table 4.4: **Change in outer diameter effect on various parameters**

O.D	H.T area	Nt	L	T.L	U
19.05	26355.06	43982	10.01	2.828	2594.32
22.225	26756.11	31580	12.14	2.617	2555.43
25.4	26756.11	23765	14.10	2.419	2555.43
28.575	27169.95	18540	16.33	2.289	2516.51
31.75	27169.95	14858	18.34	2.154	2516.51
34.925	27596.87	12190	20.63	2.063	2477.58

O.D	% A	% Nt	%L	%T.L	%U
19.05	1.52	-28.20	21.28	-7.46	-1.50
22.225	0.00	-24.75	16.14	-7.57	0.00
25.4	1.55	-21.99	15.82	-5.37	-1.52
28.575	0.00	-19.86	12.31	-5.90	0.00
31.75	1.57	-17.96	12.49	-4.22	-1.55
34.925	-	-	-	-	-

The values shown in Table and plots are only for a cooling water velocity of 1.7474 m/s.

The effect of changing the tube outer diameter from 25.4 mm to 28.575 mm on various parameters is shown in Table 4.5 :

Table 4.5: **The design parameters for two different tube outer diameters**

Particulars	**Unit**	**Value**	**Value**
Tube outer diameter	mm	25.4	28.575
Cooling water velocity	m/s	1.7474	1.7474
Heat duty	kCal/hr	300297115.9	300297115.9
LMTD	°C	4.7303	4.7303
Heat transfer coefficient	kCal/hr m^{2}°C	2555.43	2516.51
Heat transfer area	m^2	26756.0658	27169.8715
Number of tubes	-	23765	21445
Length of tubes	m	14.1	14.1
Total hydraulic loss	mWc	2.419	2.037
Shell diameter	m	5.22	5.75

The result of analysis show that as we increase the tube diameter, the velocity of cooling water flow decrease and total hydraulic loss or pressure drop also decrease drastically which increase the performance, efficiency and power output of the Thermal Power Plant. Though it will increase the required heat transfer area.

4.1.6 Tube material analysis

Knowing the limitations of material is critical when making a selection for a specific application. The properties required when selecting a material are corrosion resistance, stress corrosion cracking potential, thermal and mechanical properties, erosion resistance, vibration potential and temperature limitations[16].The property comparison guides are intended to be quick tools to assist the user in selecting a cost-effective material for specific application.

Corrosion

Corrosion may be categorized into two broad categories, general corrosion and localized corrosion which is driven by an electrochemical mechanism.

General corrosion

General corrosion is the regular dissolution of surface metal. Two most common problems faced are the rusting of carbon steel and the wall thinning of copper alloys. General corrosion is normally does not cause sudden great damage. Copper alloys are generally chosen for condensing and BOP (balance of plant) heat exchangers, and 25-year life times are not uncommon. In some applications, copper alloys are expected to slowly dissolve to maintain some resistance to bio-fouling as the copper ion can be toxic to the microorganisms that attach to the tube wall. The copper can replete on turbine blades, resulting in a loss of efficiency or tubes resulting in premature failure. So in some North American regions, high discharge levels have prevented the reuse of copper alloys in power plant condensers[16].

Electrochemically driven mechanism

These failure mechanisms usually have two modes: an incubation or an initiation mode and a propagation mode. The conductivity of several corrosion-related mechanisms are electrochemically driven and are very unpredictable. Water may be a dominant factor.

Pitting corrosion

Pitting corrosion is a highly localized attack that can result in through-wall penetration in very short duration. Failure may occur in less than 4 weeks. The most common cause of pitting of stainless steels in power industry is chlorides.

By analyzing the impact of each element, Rockel developed a formula to determine the total stainless steel resistance to chloride pitting:

$$PREn = \%Cr + 3.3(\%Mo) + 16(N) \tag{4.1}$$

PREn represents the **"Pitting Resistance Equivalent number".**

The higher PREn, the more chloride resistance and interesting to see that nickel, a common stainless steel alloying element has very small or no effect on chloride pitting resistance.

Crevice corrosion

Crevice corrosion is similar to pitting corrosion. This drives pH lower. The end result is that crevice corrosion can happen at temperature between 30 to 50 °C lower than pitting in the same condition. Crevice corrosion generally measured by the ASTM G48 test.

Stress corrosion cracking

Stress corrosion cracking is a failure mechanism that can make rapid failure when specific conditions coexist. This failure mechanism is identified from other brittle type failures such as fatigue by the branching and secondary cracking. In 300 series stainless steels, it mostly occurs in the desuperheating zone of a feedwater heater where conditions can concentrate chlorides[16]. The triangular cracking in TP 304N material Tubes due to stress corrosion is shown in fig. 4.6.

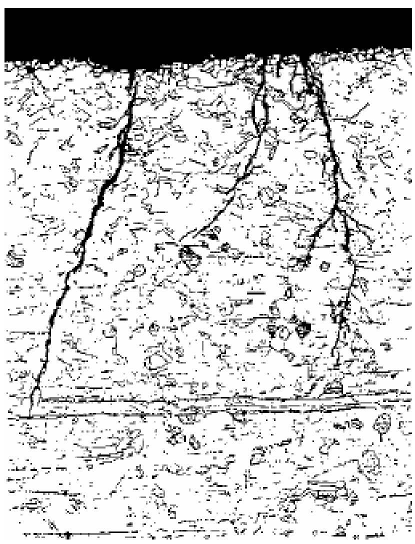


Figure 4.6: **Triangular cracking in TP 304N Tubing's[16]**

Common sources of corroding media in the power industry comprise ammonia for copper alloys and chlorides for the stainless steel alloys.

Not all stainless steels are liable to be influenced by stress corrosion cracking. The Copson curve shown in fig 4.7 exhibits a direct relationship exists between the time to failure and nickel content.

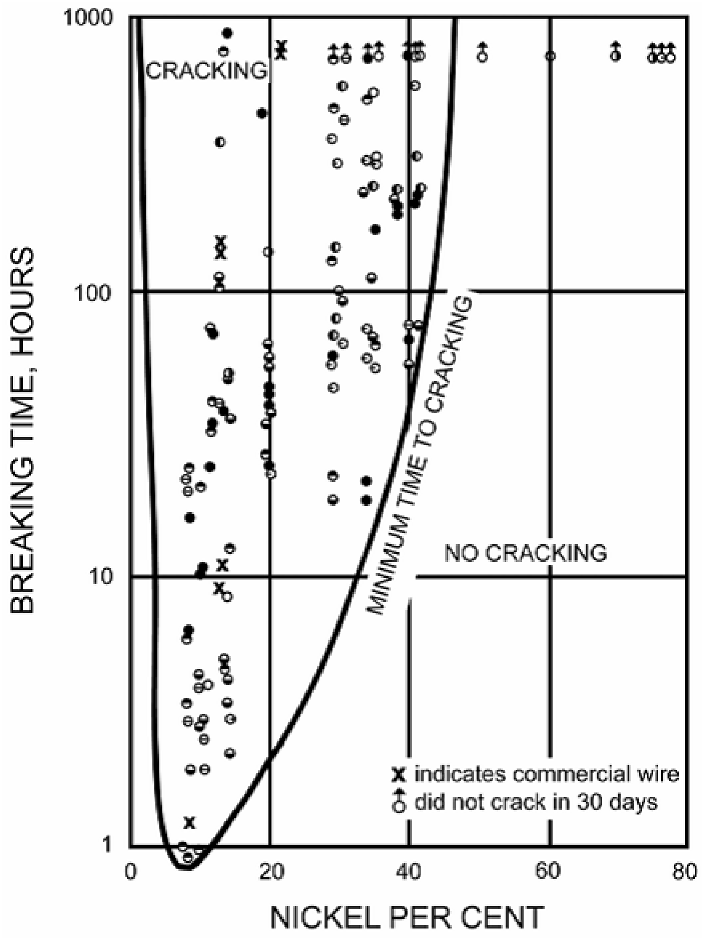


Figure 4.7: **Copson Curve**[16]

The stainless steel with nickel content of 8% is the workhorse of the industry, TP304 and TP316. Improvement in time to failure come from selecting an alloy with very low nickel content such as TP 439 or very high nickel content such as 6% molybdenum containing alloys or alloy 20. The high nickel choice can be very expensive. Mirroring the Copson curve, the alloy hold within 8% nickel failed first. Interestingly, one of the most popular choices for high pressure feedwater heater, TP304N failed rapidly than others. In this testing only TP439, the alloy containing no nickel, escaped cracking[16].

Erosion problem

Erosion resistance is a function of the ability of protective layer to remain attached to the substrate and the strength of substrate directly below the protective layer. Two types of erosion generally cause problem in the power industry – flow assisted erosion and steam impingement erosion.

Economic consideration

Prices of material can change considerably depending upon the quantity purchased, availability and OD to wall ratio. One should be very careful when assembling long-term budgets for alloys that have higher nickel contents such as TP304, TP316, cupro-nickels and 6% molybdenum containing alloys. Alloy with low nickel content like admiralty brass, TP439 and superferritics are more stable and predictable.

The calculation shows the saving of material for condenser tubes by changing from SS304 to SS439 with chemical composition and properties for both materials.
Both materials can be used by manufacturers for condenser tube manufacturing. But some time temperature, size and many other parameters can effect on performance and life of tubing.
The chemical compositions of SS 304 and SS (UNS S43035) are shown in the following Table 4.6.

Table 4.6: **Chemical properties[17]**

Sr No.	Element	SS 304	SS (UNS S43035)
1	Carbon	0.07 max	0.03 max
2	Manganese	2.00 max	1.00 max
3	Sulfur	0.03 max	0.03 max
4	Phosphorus	0.045 max	0.04 max
5	Silicon	0.75 max	1.00 max
6	Chromium	17.5 - 19.5 max	17 – 19 max
7	Nickel	8 - 10.5	0.5 max
8	Nitrogen	0.1 max	0.03 max
9	Aluminium	-	0.15 max
10	Titanium	-	[0.2 + 4*(C+N)]min -0.75 max

The mechanical and physical properties of SS 304 and SS (UNS S43035) are shown in the following Table 4.7.

Table 4.7: **Mechanical & Physical properties[17]**

Sr no.	**Element**	**SS304**	**SS439**
1	Yield strength 0.2% offser (KSI)	30 min	30 min
2	Tensile strength (KSI)	75 min	60 min
3	% elongation (2" gauge length)	40 min	22 min
4	Hardness Rockwell	HRB 92	HRB 90
5	Density (lb/in^3)	0.29	0.278
6	Modulus of elasticity in Tension ($psi*10^6$)	28	29
7	Thermal conductivity (BTU/hr*ft2*F/ft)	8.6 (70 °F)	14.0 (70 °F)
8	Mean coefficient of thermal expansion	9.2 (70 °F)	5.6 (70 °F)
9	Electrical Resistivity (micro ohm-cm)	72 (70 °F)	63 (70 °F)
10	Oxidation resistance	1650	1700

The calculation of tube material requirement for two different materials is shown in the following Table 4.8.

Table 4.8: **Tube material calculations**

Sr no.	Particulars	unit	SS 304	SS (UNS S43035)
1	Tube outer diameter	mm	25.4	25.4
2	Tube inner diameter	mm	23.98	23.98
3	Density	kg/m^3	8027.17	7695.02
4	Length	m	14.07	14.07
5	No. of tubes	-	23820	22174
6	Volume occupied by tube	m^3	0.000774275	0.000774275
7	Material occupied by tube	kg	6.22	5.96
8	Tonnage of materials occupied by tubes	Ton	148.05	132.11
9	Difference in tonnage of material required	Ton	15.935	-

The calculation show that as we change the material from SS304 to SS439 for tubes then there will be a noticeable reduction in the condenser tube material requirements and that reduction is around more than 10% of weight of tube materials.

4.1.7 Cooling Water Velocity Analysis

The circulating water velocity is the average velocity of circulating water through the tubes. Due to a change in the tube inside cooling water velocity, heat transfer area, length of tubes, pressure drop , overall heat transfer coefficient and the number of tubes also change accordingly. The changes in all parameters according to velocity are shown in the plots below:

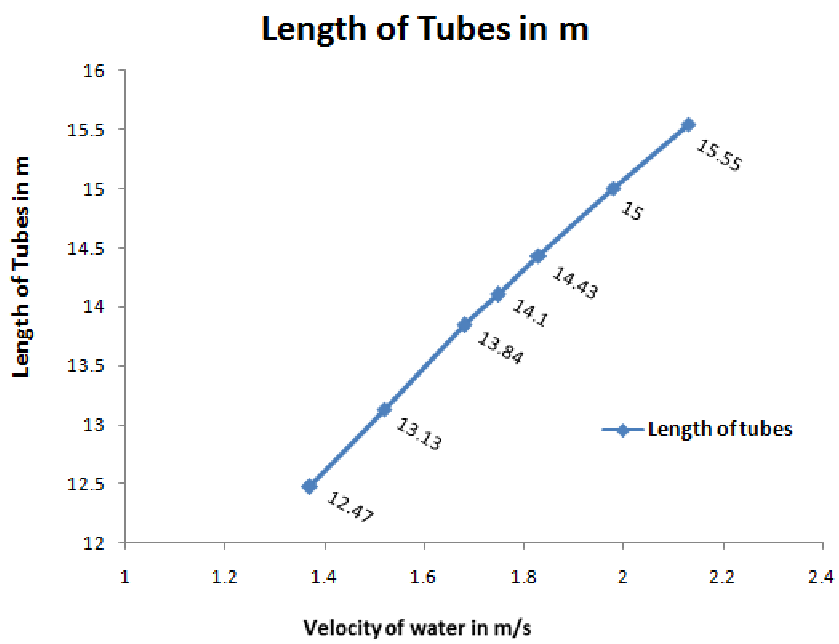


Figure 4.8: **Length of Tube v/s Cooling water Velocity**

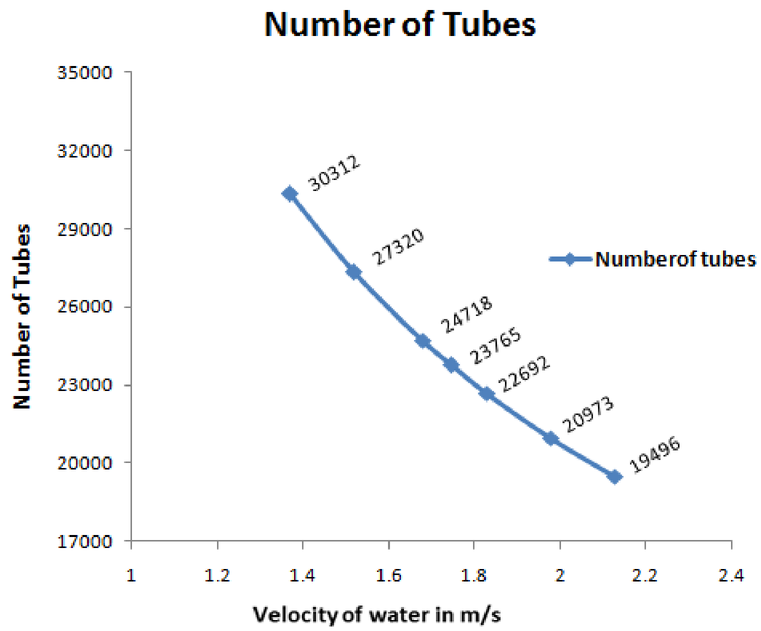


Figure 4.9: **Number of Tubes v/s Cooling water Velocity**

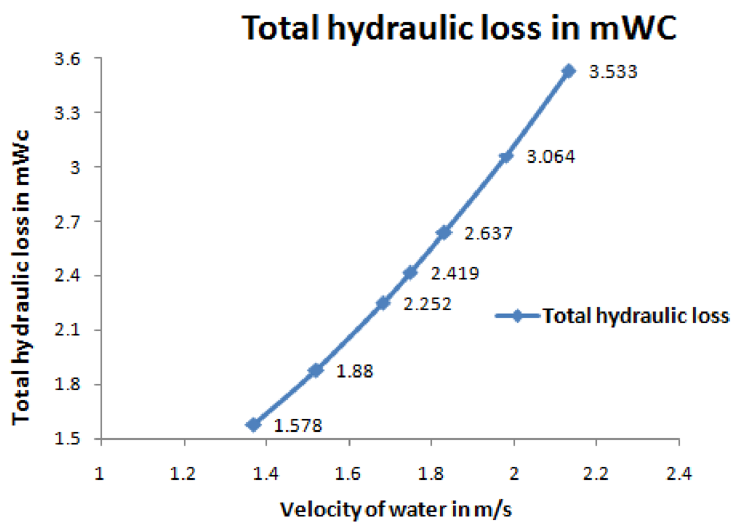


Figure 4.10: Pressure Drop v/s Cooling water Velocity

4.1.8 Exergy Analysis of Steam Surface Condenser

Exergy is a generic term for a group of concepts that define the maximum possible work potential of a system, a stream of matter or heat interaction; the state of the environment being used as the datum state. In an open flow system there are three types of energy transfer across the control surface namely working transfer, heat transfer, and energy associated with mass transfer or flow. The work transfer is equivalent to the maximum work, which can be obtained from that form of energy. The exergetic performance analysis not only determine magnitudes, location and causes of irreversibility in the plants, but also provides a more meaningful assessment of plants' individual component's efficiency.

Such a comprehensive analysis will be a more convenient approach for the performance evaluation and determination of the steps towards improvement. Exergy of a thermodynamic process shows efficiency and inefficiency of that process. Exergy provides us with a better understanding of processes for qualifying energy. Therefore, it would be better to use exergy to locate, qualify and quantify energy destruction. Exergy can play an important role in strategic development of power plants and provision of use of instruction in existing power plant. As energy analysis is based on the first law of thermodynamics, it has some inherent limitations such as accountability for the properties of the systemic degradation of energy quality through dissipative processes [6]. An energy analysis does not characterize the irreversibility of processes within a system. In contrast, exergy analysis will characterize the work potential from a system. Exergy is the maximum work that can be obtained from the system, when its state is brought to the reference or ''dead state (standard atmospheric condition). Exergy analysis is based on the second law of thermodynamics.

The detailed exergy analysis is shown in the Appendix C.

Table 4.9: Exergy Analysis of Condenser

Exergy Analysis of Condenser							
Ex = m [(h - ho) - To (S - So)]							
Particulars	m	T	h	s	h-ho	S-So	m*((h-ho)-To(s-so))
Unit	(kg/hr)	(oC)	(kJ/kg)	(kJ/kg K)	(kJ/kg)	(kJ/kg K)	(kJ/hr)
ambient (o)	-	40	167.6	0.57	0	0	0
BFPT to HP Shell	56857	45.2	2468.560	7.8	2300.960	7.23	2158860.29
BFPT to LP Shell	55083	40.7	2443.456	7.824	2275.856	7.254	294914.382
LP-2 to HP Shell	521839	43.42	2354.337	7.48	2186.737	6.91	12475500.61
LP-1 to LP Shell	521037	38.92	2331.743	7.484	2164.143	6.914	31887.4644
GC to HP Shell	41236	46.3	212.129	0.712	44.529	0.142	3414.3408
Make up	58715	41.3	172.799	0.5828	5.199	0.0128	70035.252
Condenser Outlet	1254767	41.3	172.799	0.5828	5.199	0.0128	1496435.124

Exergy in	kJ/hr	15034612.33
Exergy out	kJ/hr	1496435.124
Exergy Destruction	kJ/hr	13538177.21
	kJ/s	3760.604781

4.1.9 Result and Discussion

For the parametric analysis of the condenser, the parameters such as TTD, tube outer diameter, tube material, cooling water velocity and exergy analysis are considered.

The reference TTD value is considered as 2.838 °C on which the condenser is designed and then one lower value (2.7 °C) and one higher value (3 °C) are considered for the analysis. The analysis shown that as the TTD value increases from lower to higher, the effective heat transfer area and number of tubes are decrease for same heat duty as well as length of tube.

The tube outer diameter is 25.4 mm and then one higher size 28.575 mm is considered for the analysis. The analysis shown that as tube outer diameter increases, the required number of tubes decreases and pressure drop also decreases which will improve the performance of the system.

Tube material is a critical parameter. Because it will directly reflect in the economics of the equipment. The general industry usage tube material is stainless steel SS 304. But one more stainless steel alloy material SS 439 is used as a condenser tube material. The analysis of tube material shown that by changing the tube material from SS 304 to SS 439, the total savaing of tube material is around 15 ton which is almost 10 % of overall weight of tube material utilized. The main factor behind this saving is the thermal conductivity. Its is also shown in the property comparison Table 4.6 and Table 4.7.

The reference cooling water velocity is considered as 1.7474 m/s and for that velocity the presure drop is 2.419 mWc. According to HEI standards, the maximim velocity of cooling water is permitted upto 2 m/s for good performance. The analysis shown that as the velocity cooling water decreases from 1.7474 m/s the pressure drop decreases to a remarkable level as well as length of tube also decreases. so ultimately it will increase the performance of condenser.

The exergy analysis is performance based analysis. it depends on temperature, enthalpy, entropy and ambient working conditions. General literature review shown that for subcritical power plants condenser exergy destruction value is around 12-20 MW. One example of that is exergy analysis of 210 MW Vijayawada Thermal power station which is sub-sritical power station. The condenser exergy analysis of Vijayawada power station shown that the exergy destruction is 19 MW. The condenser exergy analysis of ultra super-critical power plant shown that the exergy destruction is only 4 MW. So there is a remarkable difference in the exergy destruction of condenser for sub-critical power plant and ultra super-critical power plant.

4.2 Parametric Analysis of Feedwater Heater

Feedwater Heaters are shell and tube heat exchangers, U-tube type heaters,which perform the special function of recovering heat from turbine extraction steam by preheating the boiler feedwater. High-pressure feedwater heaters are located downstream of the high-pressure boiler feed pump whereas low-pressure feedwater heaters are located downstream of the condensate pump. The steam that is extracted from the turbine is in most cases available at a superheated state[21].

The main objectives of feedwater heater in thermal power plants are :

1. To improve the thermodynamic efficiency of the steam cycle (so called Carnotization of the cycle)
2. To lower the thermal shock to the boiler when the feedwater enters the boiler
3. To disburden the LP Turbine Load.

4.2.1 Design Calculation of Low Pressure Feedwater Heater for a 660 MW Thermal Power Plant

The detailed design calculation of low pressure feedwater heater is shown in the Appendix B.

The input data for the design calculation of a loe pressure feedwater heater for a 660 MW Thermal power plant is provided in Table 4.10.

Table 4.10: **Input Data[25]**

Sr No.	**Particulars**		**Unit**	**Condensing Zone**		**Drain Cooling Zone**	
				Shell Side	**Tube Side**	**Shell Side**	**Tube Side**
1	Operating Presure	P	kg/cm^2 abs.	3.43	-	-	-
2	Saturation Temperature	Ts	^{o} C	137.5	-	-	-
3	Inlet Temperature	Ti	^{o} C	223.3	102.61	137.47	98.80
4	Outlet Temperature	To	^{o} C	137.47	134.70	104.4	102.61
5	Mass flow rate	M	kg/s	45.00	393.643	45.00	393.643
6	Density	r	kg/m^3	1.83	956.461	928.349	959.207
7	Thermal Conductivity	k	W/m K	0.0297891	0.679883	0.683482	0.6787
8	Dynamic Viscosity	m	kg/m s	0.0000135603	0.000274198	0.0002003990	0.000285337
9	Specific Heat	Cp	kJ/kg K	2.29192	4.21875	4.27700	4.21431
10	Inner diameter	di	mm	17.3			
11	Outer diameter	do	mm	19.1			
12	Tube thickness	t	mm	0.9			
13	Pitch	P	mm	23.813			
14	Tube material conductivity	km	W/m K	22.138			
15	Tube Fouling resistance	rt	m^2 K/ W	0.00003526			
16	Water velocity	Vw	m/s	1.855			
17	Tube material		-	SS TP 304			
18	Terminal Temperature Difference	TTD	^{o} C	2.8			
19	Shell Fouling Resistance	rs	m^2 K/ W	0.00005246			

The design calculation for the feedwater heater is shown in Table 4.11.

Table 4.11: **Calculated Result**

Sr No.	Particulars		Unit	Condensing Zone		Drain Cooling Zone	
				Shell Side	Tube Side	Shell Side	Tube Side
1	Heat duty	Q	kJ/s	53668.8170	53668.4769	6335.3524	6311.0306
2	Number of Tubes	Nt	-		944	-	941
3	Reynolds No.	Re	-	302854.7235	111941.98	30565.4517	107880.83
4	Prandtl No.	Pr	-	1.043305195	1.7014	1.254029401	1.7718
5	Friction Factor	f	-	-	0.0044	-	0.0044
6	Nusselt No.	Nu	-	-	316.1508	-	313.9816
7	Heat Transfer Coefficient	h	W/m^2 K	13567.513	12424.60	5701.180416	12317.88
8	Total flow area through tubes	At	m^2	0.2219		0.2212	
9	Tube Pitch Ratio	PR	-	1.2468		1.2468	
10	Tube Count calculation Constant	CTP	-	0.9000		0.9000	
11	Tube Layout Constant (triangular Pitch)	CL	-	0.8700		0.8700	
12	Inside Diameter	Ds	m	1.5409		1.5387	
13	Equivalent Diameter	De	m	0.0136		0.0136	
14	Baffle Spacing	B	m	0.490		0.329	
15	Clearance	C	m	0.0047		0.0047	
16	Bundle cross flow area	As	m^2	0.1494		0.1002	
17	Overall Heat transfer Coefficient	U	W/m^2 K	4157.5814		2452.8949	
18	LMTD		^oC	12.7211		16.0015	
19	Heat Transfer Area	A	m^2	1014.744		161.410	

4.2.2 Tube Mean Metal Temperature

Mean Metal Temperature :

The Mean Metal Temperature of either the shell or tubes is the temperature taken at the metal thickness midpoint averaged with respect to tube length. For the case of integrally finned tubes, the temperature at the prime tube metal thickness midpoint applies. The fin metal temperature should not be weighted with the prime tube metal temperature[26].

Fluid Average Temperature :

The shell or tube fluid average temperature is the bulk shell or tube fluid temperature averaged with respect to the exchanger tube.

Relationship between Mean Metal Temperatures and Fluid Average Temperature :

1. Shell Mean Metal Temperature :

The Shell Mean Metal Temperature generally assumed to be equal to The Shell Fluid Average Temperature.

$$T_m = T$$

This assumption is valid for cases without abnormal rates of heat transfer between the shell and surroundings. If significant heat transfer to or from the shell could occur, determination of the effect on the shell metal temperature should be made. In general, most high or low temperature externally insulated exchangers and moderate temperature non-insulated exchangers meet the above assumption.

2.Tube Mean Metal Temperature :

The Tube Mean Metal Temperature is dependent not only on the tube fluid average temperature, but also the shell fluid average temperature, the shell and tube heat transfer coefficient, shell and tube fouling resistance and Tube metal resistance to heat transfer.

According to the following relationship, the tube mean metal temperature is calculated as :

$$T_m = T - \left[\frac{\left(\frac{1}{h_o} + r_o\right) \times \left(\frac{1}{E_f}\right) + \frac{r_w}{2}}{\left(\frac{1}{h_o} + r_o\right) \times \left(\frac{1}{E_f}\right) + r_w + \left(\frac{1}{h_i} + r_i\right) \times \left(\frac{A_o}{A_i}\right)} \right] \times (T - t) \tag{4.2}$$

h_o	=	Film coefficient of shell side fluid	T_m	=	Tube mean metal temperature
h_t	=	Film coefficient of tube side fluid	T	=	Shell side fluid average temperature
r_o	=	Fouling resistance on outside surface of tubes	t	=	Tube side fluid average temperature
r_i	=	Fouling resistance on inside surface of tubes	r_w	=	Resistance of tube wall referred to outside surface of tube wall

Tube Mean Metal Temperature Calculation :

Average Shell side fluid temperature	T	°C	421
Film Coefficient on Shell side	h_s	W/ m^2 °C	430
Fouling resistance on shell side	r_s	m^2 °C/W	0.00005287
Average Tube side fluid temperature	t	°C	300
Film Coefficient on Tube side	h_t	W/ m^2 °C	15646.77
Fouling resistance on tube side	r_t	m^2 °C/W	0.00003526
OD of tube	do	mm	19.05
ID of tube	di	mm	11.938
Tube wall resistance	r_w	m^2 °C/W	0.0002
Fin efficiency	E_f	-	1
Tube mean metal Temperature	$\mathbf{T_m}$	**°F**	**593**
		°C	**312**
Tube Design Temperature		**°C**	**325**

4.2.3 TTD Analysis of Feedwater Heater

The performance of a feedwater heater depends on many parameters but mainly affecting parameters are Terminal Temperature Difference (TTD), Drain Cooling Approach (DCA), Temperature Rise (TR).

In the present study, Terminal Temperature Difference (TTD) is considered for analyzing the performance of feedwater Heater[6]. For both LP Heater and Hp Heater, TTD analysis performed is as following :

Low Pressure Heater (LP Heater) :

Generally LP Heater is a two zone heater. One is the Condensing Zone and the other is the Drain Cooling Zone. In most cases there is no availability of Desuperheating Zone. So generally the TTD value for LP Heater is **POSITIVE.**

The change in the effective heat transfer area by changing the TTD values for LP feedwater heaters are tabulated in Table 4.12 and behaviour of curve accoding to the values is shown in fig. 4.11.

Table 4.12: **TTD Analysis of LP Heater**

Effective Heat transfer Area in m^2 (U = Constant)			
TTD (°C)	**Condensing Zone**	**Drain Cooling Zone**	**Total Area (m^2)**
2.0	357.044	10.144	367.188
2.1	349.490	10.012	359.502
2.2	342.342	9.883	352.225
2.3	335.566	9.758	345.324
2.4	329.126	9.636	338.762
2.5	322.997	9.517	332.514
2.6	317.151	9.402	326.553
2.7	311.567	9.289	320.856
2.8	306.226	9.179	315.405

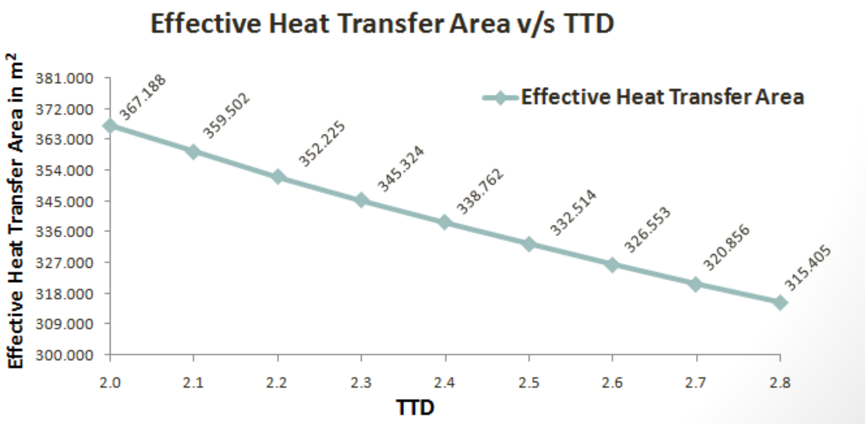


Figure 4.11: **Effective heat transfer area of LP heater v/s TTD**

High Pressure Heater (HP Heater) :

Generally HP Heater is a three zone heater. The Desuperheating Zone is included in HP Heater with Condensing Zone and Drain Cooling Zone. So generally the TTD value for HP Heater is **NEGATIVE**. Because of Desuperheating Zone the feedwater outlet temperature becomes higher than saturation temperature of steam[27].

The change in the effective heat transfer area by changing the TTD values for HP feedwater heaters are tabulated in Table 4.13 and behaviour of curve accoding to the values is shown in fig. 4.12.

Table 4.13: **TTD Analysis of HP Heater**

Effective Heat transfer Area (U = Constant)					
TTD (°C)	**DS1**	**DS2**	**Cond**	**DC**	**Total Area (m^2)**
-2.6	72.684	72.923	2023.896	257.237	2426.740
-2.5	72.518	72.763	1771.022	254.312	2170.615
-2.4	72.354	72.603	1622.244	251.467	2018.668
-2.3	72.191	72.444	1516.785	248.697	1910.116
-2.2	72.029	72.285	1435.220	245.999	1825.533
-2.07	71.819	72.081	1351.099	242.594	1737.593
-1.9	71.547	71.816	1264.683	238.307	1646.353
-1.8	71.388	71.661	1222.331	235.867	1601.248
-1.7	71.230	71.507	1184.631	233.486	1560.854
-1.6	71.073	71.354	1150.646	231.160	1524.234
-1.5	70.917	71.202	1119.776	228.888	1490.783

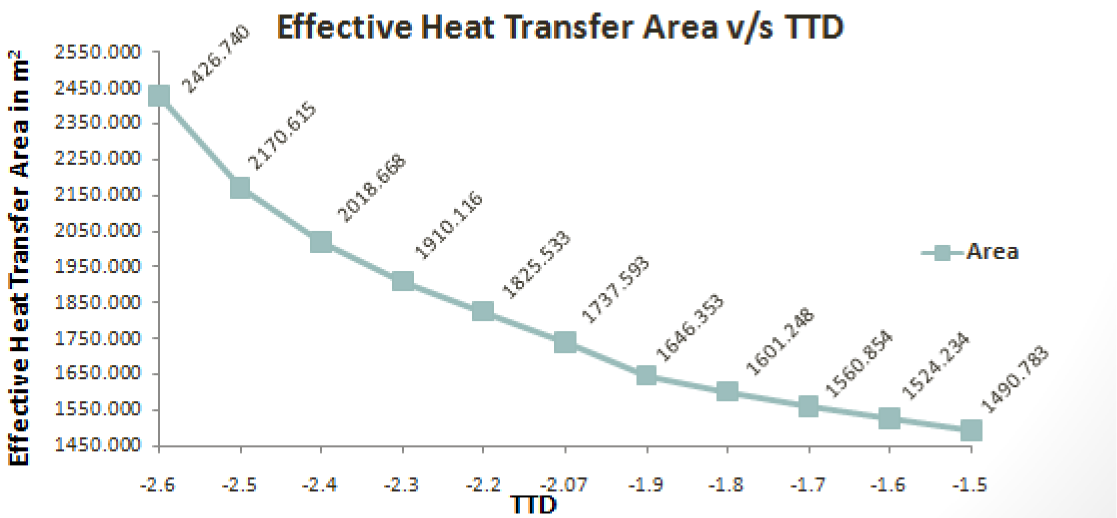


Figure 4.12: **Effective heat transfer area of HP heater v/s TTD**

4.2.4 TTD analysis of LP Heater without and with Desuperheating Zone

TTD Analysis of LP Heater Without Desuperheating Zone:

LP Heater effective heat transfer area analysis without desuperheating zone at different TTD values are shown as below :

The change in the effective heat transfer area by changing the TTD values for LP feedwater heaters without Desuperheating Zone are tabulated in Table 4.14 and behaviour of curve accoding to the values is shown in fig. 4.13.

Table 4.14: **TTD Analysis of LP Heater without Desuperheating Zone**

Effective Heat transfer Area (U = Constant) in m^2			
TTD (°C)	**Condensing Zone**	**Drain Cooling Zone**	**Total Area (m^2)**
2.0	1135.232	174.199	1309.431
2.1	1117.344	172.627	1289.971
2.2	1100.345	171.091	1271.436
2.3	1084.155	169.590	1253.745
2.4	1068.705	168.123	1236.828
2.5	1053.933	166.687	1220.620
2.6	1039.786	165.283	1205.069
2.7	1026.216	163.908	1190.124
2.8	1013.182	162.562	1175.744

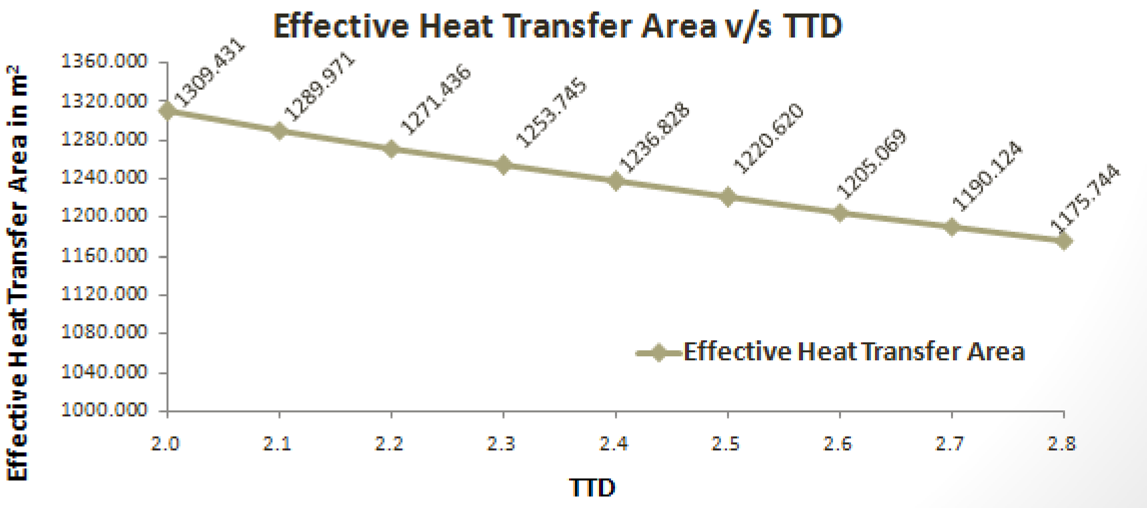


Figure 4.13: **Effective heat transfer area without Desuperheating Zone v/s TTD**

TTD Analysis of LP Heater with Desuperheating Zone :

LP Heater effective heat transfer area analysis with Desuperheating Zone at different TTD values are shown as below :

The change in the effective heat transfer area by changing the TTD values for LP feedwater heaters with Desuperheating Zone are tabulated in Table 4.15 and behaviour of curve accoding to the values is shown in fig. 4.14.

Figure 4.14: **TTD Analysis of LP Heater with Desuperheating Zone**

Effective Heat transfer Area (U = Constant) in m^2				
TTD (°C)	Desuperheating Zone	Condensing Zone	Drain Cooling Zone	Total Area
2.0	196.159	811.765	173.975	1181.899
2.1	194.778	804.449	172.406	1171.633
2.2	193.428	797.311	170.872	1161.611
2.3	192.108	790.345	169.374	1151.827
2.4	190.818	783.542	167.909	1142.269
2.5	189.554	776.896	166.476	1132.926
2.6	188.318	770.402	165.073	1123.793
2.7	187.107	764.051	163.701	1114.859
2.8	185.920	757.840	162.357	1106.117

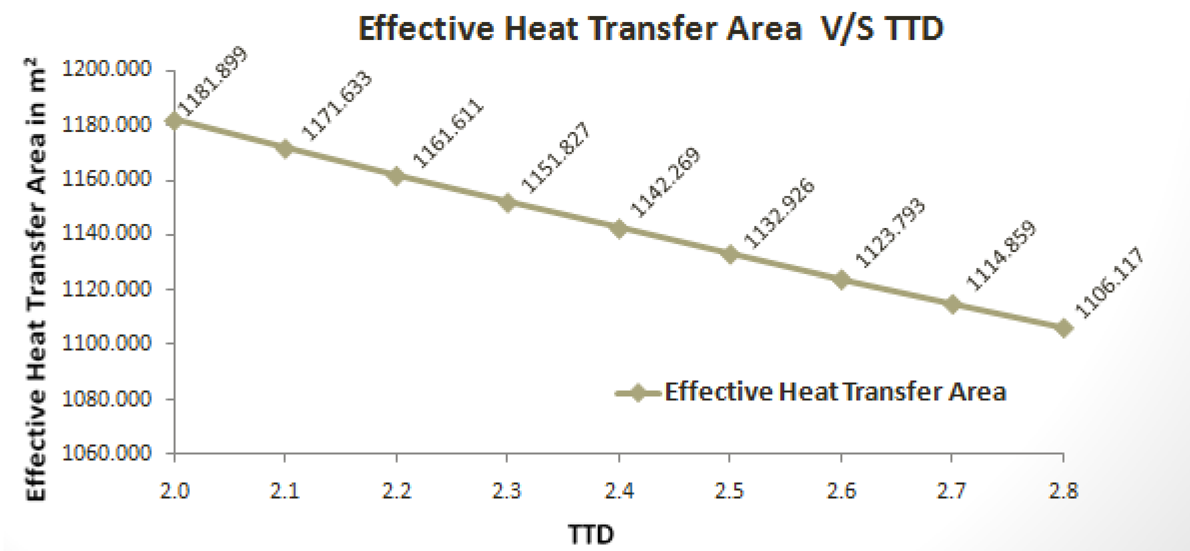


Figure 4.15: **Effective heat transfer area with Desuperheating Zone v/s TTD**

4.2.5 Results and Discussion

For the analysis of feedwater heater, the parameter TTD for LP heater and HP heater and desuperheating zone addition in the LP heater is considered.

The TTD analysis for LP heater is shown in Fig. 4.11. Generally the TTD values for LP heater are positive and as the TTD value increases, the total heat transfer area decreases linearly.

The TTD analysis for HP heater is shown in Fig. 4.12. Generally the TTD values for HP heater are negative however because of the desuperheating zone provided and as the TTD value increases, the total heat transfer area decreases exponentially.

If the superheat is available with steam at inlet of LP heater and a Desuperheating zone is provided then the total area of heater decreases at a renmarkable level as compared to that without a Desuperheating Zone. In the Desuperheating Zone the heat transfer is single phase heat transfer whereas in the condensing zone the heat transfer is two phase heat transfer. So without Desuperheating Zone there is a maximum heat transfer in the condensing zone which require large area for heat transfer due to two phase condition.

Chapter 5

Conclusions

5.1 Conclusions

The present study is conducted on Heat Exchangers considering the following parameters:

Condenser :

i. TTD
ii. Tube outer diameter
iii. Tube material
iv.Velocity of Cooling water
v. Exergy Analysis

The following conclusions are drawn from the present study :
i. In the prevalent design calculations, a fix value of TTD is being considered for design (TTD = 2.838 °C). In case the value of TTD is increased and the heat duty as well as the length of tube is kept constant, all the other parameters such as the number of tubes and the effective heat transfer area decrease.

ii. As the outer tube diameter increases, the heat transfer area and the length of tube increases although the number of tubes and the main parameters such as the total hydraulic loss or pressure drop decreases to a remarkable level. Hence it improves the overall performance of the system.

iii. In the present design, SS304 is being used as a tube material in the heat exchanger construction. The thermal design calculations considering SS439 as a tube material instead of SS304 reveal that in case SS439 is used, there is a saving of about 10% in the overall weight of tube material utilized.

iv. In the prevalent design calculations, a fix value of velocity of cooling water is being considered as 1.7474 m/s in the heat exchanger. In case the velocity of cooling water decreases, the pressure drop inside the cooling water tubes decreases at remarkable level as well as length of tubes also decreases but number of tubes increases.

v. In the exergy analysis of condenser, it is found that exergy destruction in the condenser is 3760 KW which is much lesser than other condenser of sub-critical power plants because of working pressure difference.

For the feedwater heater the following parameters were studied :

i. TTD of LP and HP Heater
ii. Desuperheating Zone addition in LP Heater

The following conclusions are drawn from the present study :

i. In the prevalent design calculation, a fix value of TTD is being considered for design. In case the value of TTD increases, the total area of heater linearly decreases for LP Heater where as the total area of heater exponentially decreases for HP Heater.

ii. A comparison of the LP Heater without Desuperheating Zone and with Desuperheating Zone shows that if superheat is available with steam and the Desuperheating Zone is provided then the total area of heater decreases at a remarkable level as compared to that without the Desuperheating Zone.

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Appendix A : Design Calculation of surface condenser

The input data for the design of steam surface condenser is provided as following :

1.	Condenser operating pressure (P_s)	=	0.0696 bar
2.	Steam saturation Temperature (T_s)	=	38.92 °C
3.	Terminal Temperature Difference (TTD)	=	2.838 °C
4.	Cooling water inlet temperature (T_1)	=	31.60 °C
5.	Specific heat of Cooling water (C_p)	=	0.99803 kCal / kg °C
6.	Tube outer diameter (d_o)	=	25.4 mm
7.	Thickness of tube @ 22 BWG	=	0.711 mm
8.	Specific volume of Cooling water (V_f)	=	0.001005553 m^3 / kg
9.	Tube water Velocity	=	1.7474 m / s
10.	Tube material	=	SS 304
11.	Cooling water type	=	Fresh water
12.	Cleanliness factor (F_c)	=	0.9

Input for Heat load and Cooling water flow calculation

1.	Steam inlet from main turbine exhaust (m_1)	=	521037 kg / hr
	Enthalpy (h_1)	=	557.3 kCal / kg
2.	Steam inlet from BFP drive turbine (m_2)	=	55083 kg / hr
	Enthalpy (h_2)	=	584 kCal / hr
3.	Feedwater heater drains	=	19528 kg/ hr
	Enthalpy (h_3)	=	48 kCal / hr
4.	Enthalpy for Condensate outlet	=	38.92 kCal / hr

Heat load calculation :

The heat load calculation is done as following :

$$Q_{input} = \sum m_n \times h_n = (m_1 \times h_1) + (m_2 \times h_2) + (m_3 \times h_3)$$

$$Q_{input} = (521037 \times 557.3) + (55083 \times 584) + (19528 \times 48)$$

$$Q_{input} = 323479736.1\, kCal/hr$$

$$Q_{output} = (\sum m_n) \times h_o$$

$$Q_{output} = (521037 + 55083 + 19528) \times 38.92$$

$$Q_{output} = 23182620.16\, kCal/hr$$

$$Q_{net} = Q_{input} - Q_{output}$$

$$Q_{net} = 323479736.1 - 23182620.16$$

$$Q_{net} = 300297115.9\, kCal/hr$$

Cooling water flow calculation :
The required cooling water flow calculation is done as following :

1. Cooling water outlet temperature

$$T_2 = T_s - TTD$$

$$T_2 = 38.92 - 2.838$$

$$T_2 = 36.082\,^oC$$

2. Maximum Temperature rise

$$TR = T_2 - T_1$$

$$TR = 36.082 - 31.6$$

$$TR = 4.482\,^oC$$

3.Cooling water flow

$$m = \frac{Q_{net}}{C_p \times TR}$$

$$m = \frac{300297115.94}{0.99803 \times 4.482}$$

$$m = 67132947.12\,kg/hr$$

4.Cooling water volumetric flow

$$m_1 = m \times V_f$$

$$m_1 = 67132947.12 \times 0.001005553$$

$$m_1 = 67505.74\,m^3/hr$$

Heat transfer surface area calculation :
The heat transfer surface area calculation is done as following :
1. Overall heat transfer coefficient

$$U = U_1 \times F_m \times F_c \times F_w$$

$$U = 3574.86 \times 0.862 \times 0.9 \times 1.0716$$

$$U = 2555.43\,kCal/hr\,m^{2o}C$$

where,

U_1= Uncorrected overall heat transfer coefficient which is taken from HEI standard at Cooling water velocity 1.7474 m/s and Tube O. D. 25.4mm.

F_m= Material correction factor which is taken from the HEI standard for tube material SS 304 and thickness 0.711 mm.

F_w= Inlet water temperature correction factor which is taken from HEI standard for Cooling water inlet temperature 31.6 °C.

F_c= Cleanliness factor generally consider between 0.75-0.9 according to condition of cooling water flow.

2. Log Mean Temperature Difference

$$LMTD = \frac{T_2 - T_1}{ln\left(\frac{T_s - T_1}{T_s - T_2}\right)}$$

$$LMTD = \frac{36.082 - 31.6}{ln\left(\frac{38.92 - 31.6}{38.92 - 36.082}\right)}$$

$$LMTD = 4.7303\, ^oC$$

3.Heat transfer surface area

$$A = \frac{Q}{U \times LMTD}$$

$$A = \frac{300297115.94}{2555.43 \times 4.7303}$$

$$A = 24843\, m^2$$

According to L&T guidelines, the tolerance value for condenser heat transfer area is 7.72 %.

So considering tolerance value, the final heat transfer area is calculated as following :

$$A_f = A + tolerance$$

$$A_f = 24843 + (24843 \times 0.0772)$$

$$A_f = 26756.11\, m^2$$

Number of tubes and Length of tube calculation :

1.Number of tubes

$$N_t = \frac{m_1}{V_w \times Volumetric\, flow\, per\, tube\, in\, m/s}$$

$$N_t = \frac{67505.74}{1.7474 \times 1.6256}$$

$$N_t = 23765$$

2.Length of tube

$$L = \frac{A_f}{\pi \times d_o \times N_t}$$

$$L = \frac{26756.11}{\pi \times 0.0254 \times 23765}$$

$$L = 14.12\, m$$

Total hydraulic loss / Pressure drop calculation :

The pressure drop calculation is done as following :

$$R_{tt} = (R_1 \times R_2 \times R_t \times L) + R_{iw} + R_{ow} + R_{te}$$

$$\left\{ R_2 \times R_t = \frac{0.00642 \times \left(\frac{V_w}{12 \times d_o}\right)^{1.75}}{\left(\frac{d_t}{d_o}\right)^{1.25}} \right\}$$

$$R_{tt} = (0.9625 \times 0.1468 \times 14.12) + 0.1899 + 0.0601 + 0.1771$$

$$R_{tt} = 2.419\, mWc$$

R_1 = Temperature factor = 0.9625 from graph given in HEI
R_{iw} = Water box inlet loss = 0.1899 from graph given in HEI
R_2 = Tube O.D. and Gauge factor
R_{ow} = Water box outlet loss = 0.0601 from graph given in HEI
R_t = Tube friction loss
R_{te} = Tube loss (Inlet + Outlet) = 0.1771 from graph given in HEI
d_t = average tube diameter

Appendix B : Design calculation for Feedwater Heater

The input data for the dsign of feedwater heater is provided as following :

Shell side input data

1.	Operating pressure (P_s)	=	3.367 bar
2.	Steam saturation temperature (T_s)	=	137.5 °C
3.	Steam flow rate (M_s)	=	82385 kg/hr
4.	Steam inlet temperature (T_{si})	=	223.3 °C
5.	Steam inlet enthalpy (h_{si})	=	695.6 kCal / kg
6.	drain inlet temperature for DC zone (T_{co})	=	137.47 °C
7.	Drain inlet enthalpy for drain cooling Zone (h_{co})	=	138.13 kCal / kg
8.	Drain outlet temperature (T_o)	=	104.4 °C
9.	Drain outlet enthalpy (h_o)	=	104.5 kCal / kg
10.	Cascade inlet drain flow rate (M_d)	=	79627 kg/hr
11.	Cascade inlet drain enthalpy (h_d)	=	141 kCal / kg

Tube side input data

1.	Feed water flow rate (M_f)	=	1417114 kg / hr
2.	Feed water inlet temperature (T_{fi})	=	98.8 °C
3.	Feed water inlet enthalpy (h_{fi})	=	98.9 kCal / kg
4.	Feed water outlet temperature for DC zone (T_{fco})	=	102.61 °C
5.	Feed water Outlet enthalpy for DC zone (h_{fco})	=	102.73 kCal/kg
6.	Feed water outlet temperature (T_{fo})	=	134.7 °C
7.	Feed water outlet enthalpy (h_{fo})	=	135.3 kCal / kg

1.	Inner diameter (d_i)	=	17.3 mm
2.	Outer diameter (d_o)	=	19.1 mm
3.	Tube thickness (t)	=	0.9 mm
4.	Pitch (P)	=	23.813 mm
5.	Tube material conductivity (k_m)	=	22.138 w / m K
6.	Tube fouling resistance (r_t)	=	0.00003526 m^2 k / w
7.	Water velocity (V_w)	=	1.855 m / s
8.	Tube material	=	SS 304
9.	Terminal temperature difference (TTD)	=	2.8 °C
10.	Shell fouling resistance (r_s)	=	0.00005246 m^2 k / w

Condensing zone calculation :

Properties are as following :

Particulars	**Unit**	**Shell side**	**Tube side**
Inlet temperature (T_i)	°C	223.3	102.61
Outlet temperature (T_o)	°C	137.47	134.70
Density (r)	kg / m^3	1.83	956.461
Thermal Conductivity (k)	w / m K	0.0297891	0.679883
Dynamic Viscosity (m)	kg / m s	0.0000135603	0.000274198
Specific heat (C_p)	kJ / kg K	2.29192	4.21875

1.Heat duty calculation

$$Q_{shell} = (m_s \times (h_{si} - h_{co}) + m_d \times (h_d - h_{co}))$$

$$Q_{shell} = \left((82385 \times (695.6 - 138.13) + (79627 \times (141 - 138.13))) \times \tfrac{4.186}{3600}\right)$$

$$Q_{shell} = 53668.4769\, kJ/s$$

$$Q_{tube} = (m_f \times (h_{fo} - h_{fco}))$$

$$Q_{tube} = \left((1417114 \times (135.3 - 102.73))\, \tfrac{4.186}{3600}\right)$$

$$Q_{tube} = 53668.477\, kJ/s$$

(A) Thermal analysis for Tube side

1. Number of tubes

$$m_t = \rho_t \times V_w \times A_c \times N_t$$

$$53668.477 \times 4 = 956.461 \times 1.855 \times \pi \times 0.0173 \times N_t$$

$$N_t = 944$$

2. Reynolds number & Prandtl number

$$Re_t = \tfrac{\rho_t \times V_w \times d_i}{\mu_t}$$

$$Re_t = \tfrac{956.461 \times 1.855 \times 0.0173}{0.000274198}$$

$$Re_t = 111941.98$$

$$Pr_t = \tfrac{\mu_t \times C_{pt}}{k_t}$$

$$Pr_t = \tfrac{0.000274198 \times 4.248}{0.679883}$$

$$Pr_t = 1.7132$$

3. Nusselt number

$$Nu_t = \frac{\left(\frac{f}{2}\right) \times Re_t \times Pr_t}{1.07 + \left(12.7 \times \left(\frac{f}{2}\right)^{1/2} \times \left(Pr_t^{2/3} - 1\right)\right)}$$

$$\{\, f = (1.58\, ln\, (Re) - 3.28)^{-2} \}$$

$$\left\{\, f = (1.58\, ln\, (111941.98) - 3.28)^{-2} \right\}$$

$$\{\, f = 0.0044 \,\}$$

$$Nu_t = \frac{\left(\frac{0.0044}{2}\right) \times 111941.98 \times 1.7132}{1.07 + \left(12.7 \times \left(\frac{0.0044}{2}\right)^{1/2} \times \left(1.7132^{2/3} - 1\right)\right)}$$

$$Nu_t = 317.4033$$

4. heat transfer coefficient

$$h_t = Nu_t \times \frac{k_t}{d_i}$$

$$h_t = 317.4033 \times \frac{0.679883}{0.0173}$$

$$h_t = 12473.82\, w/m^2k$$

(B) Thermal analysis for Shell side

1. Total flow area through tubes

$$A_t = \frac{\pi \times d_i^2 \times N_t}{4}$$

$$A_t = \frac{\pi \times (0.0173)^2 \times 944}{4}$$

$$A_t = 0.2219\, m^2$$

2. Shell inside diameter

$$D_s = 0.637 \times \sqrt{\frac{C_L}{CTP}} \times (\pi \times P_t^2 \times N_t)^{1/2}$$

$$C_L = 0.87\, For\, Triangular\, Pitch$$

$$CTP = 0.90\, For\, two - tube\, pass$$

$$D_s = 0.637 \times \sqrt{\frac{0.87}{0.9}} \times \left(\pi \times (0.023813)^2 \times 944\right)^{1/2} \times 2$$

$$D_s = 1.5409\, m$$

3. Shell equivalent

$$D_e = \frac{4 \times \left(\left(\frac{\sqrt{3}}{4} \times P_t^2\right) - \left(\frac{\pi \times d_o^2}{8}\right)\right)}{\frac{\pi \times d_o}{2}}\ For\, Triangular\, Pitch$$

$$D_e = \frac{4 \times \left(\left(\frac{\sqrt{3}}{4} \times (0.023813)^2\right) - \left(\frac{\pi \times (0.0191)^2}{8}\right)\right)}{\frac{\pi \times (0.0191)^2}{2}}$$

$$D_e = 0.0136\, m$$

4. Shell side Reynolds number

$$Re_s = \left(\frac{m_h}{A_s}\right) \times \frac{D_e}{\mu_h}$$

$$\left\{A_s = \frac{D_s \times C \times B}{P_t}\right\}$$

$$\{C = Clearance = 0.0047\, m\}$$

$$\{B = Baffle\, Spacing = 0.49\, m\}$$

$$\left\{A_s = \frac{1.5409 \times 0.0047 \times 0.49}{0.023813}\right\}$$

$$\{A_s = 0.1494\, m\}$$

$$Re_s = \left(\frac{45}{0.1494}\right) \times \frac{0.0136}{0.0000135603}$$

$$Re_s = 302854.7235$$

5. Condensing Zone heat transfer Coefficient

$$h_{cond} = 0.725 \times \left\{ \frac{k_t^3 \times \rho_t \times (\rho_t - \rho_s) \times g \times h_{fg}}{\mu \times (T_{sat} - T_{wall}) \times d_o} \right\}^{1/4}$$

$$\left\{ T_{wall} = \frac{T_{sat} + \frac{T_{co} + T_{ci}}{2}}{2} \right\}$$

$$\left\{ T_{wall} = \frac{137.5 + \frac{102.61 + 134.7}{2}}{2} \right\}$$

$$\{ T_{wall} = 128.07\,^{o}C \}$$

$$h_{cond} = 0.725 \times \left\{ \frac{(0.679883)^3 \times 956.461 \times (956.461 - 1.83) \times 9.81 \times 2150 \times 1000}{0.0000135603 \times (137.5 - 128.07) \times 0.0191} \right\}^{1/4}$$

$$h_{cond} = 13567.096\, w/m^2K$$

6. Overall heat transfer Coefficient

$$U_f = \left(\frac{1}{h_{cond}} + \frac{d_o}{d_i \times h_i} + R_{ft} + \left(\frac{d_o}{2} \times \frac{ln \frac{d_o}{d_i}}{k_t} \right) \right)^{-1}$$

$$U_f = \left(\frac{1}{13567.094} + \frac{0.0191}{0.0173 \times 12473.82} + 0.00003526 + \left(\frac{0.0191}{2} \times \frac{ln \frac{0.0193}{0.0173}}{22.183} \right) \right)$$

$$U_f = 4163.612\, w/m^2k$$

Drain cooling zone calculation :

The calculation procedure for drain cooling zone is same as mentioned for condensing zone but the change in only the equation of shell side heat transfer coefficient. Because in the condensing zone the flow behaviour is two phase and in the drain cooling zone the flow behaviour is only single phase.

For drain cooling zone the shell side heat transfer coefficient is calculated as following :

$$h_{drain} = \frac{0.36 \times k_h}{D_e} \times Re_s^{0.55} \times Pr_h^{1/3}$$

Appendix C : Exergy analysis calculation for condenser

The detail exergy analysis is calculated as following :
The ambient condition at the site of equipment working is given as following :

Temperature (T_0)	=	40 °C
Enthalpy (h_0)	=	167.6 kJ / kg
Entropy (s_0)	=	0.57 kJ / kg K

1. BFPT to HP shell steam inlet

Mass flow rate (m_1)	=	56857 kg / hr
Temperature (T_1)	=	45.2 °C
Enthalpy (h_1)	=	2468.56 kJ / kg
Entropy (s_1)	=	7.8 kJ / kg K

$$Ex_1 = m_1 \times ((h_1 - h_0) - T_0 (s_1 - s_0))$$

$$Ex_1 = 56857 \times ((2468.56 - 167.6) - 313 (7.8 - 0.57))$$

$$Ex_1 = 2158860.29 \, kJ/hr$$

2. BFPT to LP shell steam inlet

Mass flow rate (m_2)	=	55083 kg / hr
Temperature (T_2)	=	40.7 °C
Enthalpy (h_2)	=	2443.456 kJ / kg
Entropy (s_2)	=	7.824

$$Ex_2 = m_2 \times ((h_2 - h_0) - T_0 (s_2 - s_0))$$

$$Ex_2 = 55083 \times ((2443.456 - 167.6) - 313 (7.824 - 0.57))$$

$$Ex_2 = 294914.382 \, kJ/hr$$

3. LP turbine-1 to LP shell steam inlet

Mass flow rate (m_3)	=	521037 kg / hr
Temperature (T_3)	=	38.92 °C
Enthalpy (h_3)	=	2331.743 kJ / kg
Entropy (s_3)	=	7.484 kJ / kg K

$$Ex_3 = m_3 \times ((h_3 - h_0) - T_0 (s_3 - s_0))$$

$$Ex_3 = 521037 \times ((2331.743 - 167.6) - 313 (7.484 - 0.57))$$

$$Ex_3 = 31887.4644 \, kJ/hr$$

4. LP turbine-2 to HP shell steam inlet

Mass flow rate (m_4)	=	521839 kg / hr
Temperature (T_4)	=	43.42 °C
Enthalpy (h_4)	=	2354.337 kJ / kg
Entropy (s_4)	=	7.48 kJ / kg K

$$Ex_4 = m_4 \times ((h_4 - h_0) - T_0\,(s_4 - s_0))$$

$$Ex_4 = 521839 \times ((2354.337 - 167.6) - 313\,(7.48 - 0.57))$$

$$Ex_4 = 12475500.61\,kJ/hr$$

5. Gland condenser to HP shell drain inlet

Mass flow rate (m_5)	=	41236 kg / hr
Temperature (T_5)	=	46.3 °C
Enthalpy (h_5)	=	212.129 kJ / kg
Entropy (s_5)	=	0.712 kJ / kg K

$$Ex_5 = m_5 \times ((h_5 - h_0) - T_0\,(s_5 - s_0))$$

$$Ex_5 = 41236 \times ((212.129 - 167.6) - 313\,(0.712 - 0.57))$$

$$Ex_5 = 3414.3408\,kJ/hr$$

6. Make up

Mass flow rate (m_6)	=	58715 kg / hr
Temperature (T_6)	=	41.3 °C
Enthalpy (h_6)	=	172.799 kJ / kg
Entropy (s_6)	=	0.5828 kJ / kg K

$$Ex_6 = m_6 \times ((h_6 - h_0) - T_0\,(s_6 - s_0))$$

$$Ex_6 = 58715 \times ((172.799 - 167.6) - 313\,(0.5828 - 0.57))$$

$$Ex_6 = 70035.252\,kJ/hr$$

$$Ex_{in} = Ex_1 + Ex_2 + Ex_3 + Ex_4 + Ex_5 + Ex_6$$

$$Ex_{in} = 2158860.29 + 294914.382 + 12475500.61 + 31887.4644 + 3414.3408 + 70035.252$$

$$Ex_{in} = 15034612.33\,kJ/hr$$

7. Condenser outlet

Mass flow rate (m_{out})	=	1254767 kg / hr
Temperature (T_{out})	=	41.3 °C
Enthalpy (h_{out})	=	172.799 kJ / kg
Entropy (s_{out})	=	0.5828 kJ / kg K

$$Ex_{out} = m_{out} \times ((h_{out} - h_0) - T_0\,(s_{out} - s_0))$$

$$Ex_{out} = 1254767 \times ((172.799 - 167.6) - 313\,(0.5828 - 0.57))$$

$$Ex_{out} = 1496435.124\,kJ/hr$$

$$Ex_{destruction} = Ex_{in} - Ex_{out}$$

$$Ex_{destruction} = 15034612.33 - 1496435.124$$

$$Ex_{destruction} = 13538177.21\,kJ/hr$$

$$Ex_{destruction} = 3760.604781\,kJ/s$$