6

Thermal Power Station (Thermodynamic Process)

6.1. Overview

A conventional thermal power station uses fossil fuel, such as; coal, oil or gas to produce the necessary heat for the conversion of water into steam in the boiler. The superheated steam is allowed to produce kinetic energy in the turbine, which in turn rotates the generator for the production of electrical power. A simple schematic arrangement is shown in figure 6.1. The detail working of different components of thermal power station will be discussed in chapter 7.

As shown in figure 6.1, in a thermal power plant, fuel such as; natural gas, oil or coal is burnt in the combustion chamber of a furnace to produce heat. The heat is given to a steam generator or a boiler in which water is converted into superheated steam at high pressure. The superheated, high pressure steam is conveyed via a steam valve to a single or multi-stage steam turbine system. The turbine shaft is directly mechanically coupled to an electric generator, which must rotate at fixed speed to produce voltage at a fixed power frequency. The necessary speed is maintained through speed control and governing mechanism by making appropriate steam valve settings at the turbine input. In this way, electrical energy is produced and is then transmitted over transmission lines to load centers. After driving the turbine, the exhaust steam from the low-pressure turbine is passed into the condenser where hot steam is condensed into water to be reused in the boiler. Thermal power plants are outstanding because of their high efficiency and capacity and long service life. In an inter-connected system, thermal power plants are usually coordinated to run continuously all the time as base load plants for the load system.

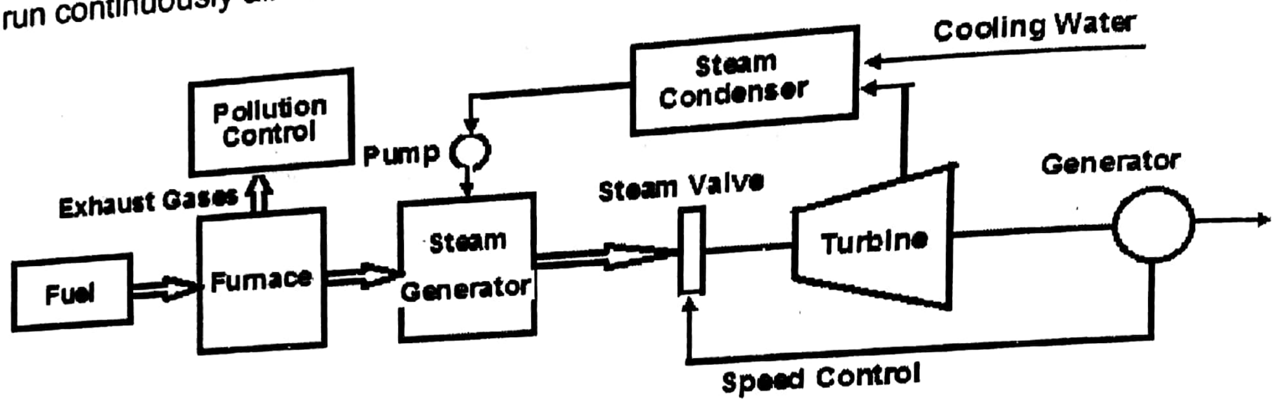


Figure 6.1: Simple Schematic Diagram of a Thermal Power Station

However, the conventional steam turbine generating plants use to supply the grid's base load are viable only as large-scale installations requiring special precautions because of their large physical size and the high voltages and currents involved. Smaller scale systems use a variety of alternative generation schemes, which can also be tailored for domestic use. Paradoxically, small-scale thermal-electric generating plant can be more complex than the large thermal-electric power plant. Coal is the most economical option to be used as a fuel despite having environmental hazards. The most controversial part of the use of coal in thermal power station is the environmental pollution and health hazards as already discussed in chapter 5. Non-practicing of proper safety procedures of using coal in thermal power stations raises voice of concern amongst the general public. Environment and health hazards are one of the most prominent reasons why many groups are against the use of coal for power production. Excessive toxic gas and fumes emission from thermal power plants adds to environment. Pollution control of emission by treatment can lower the risks of health hazards and reduce environmental pollution. Another major factor to be kept in mind is that despite the abundant supply of coal, it is still a non-renewable source of energy which is formed through a complex process lasting thousands of years and hence cannot be formulated at a short notice. Energy is available from many sources; most of the energy used to satisfy the world's ever increasing demand for electricity is still derived from fossil fuels. The dependency on fossil fuels, however, brings with it two problems.

1. Finite supplies of fossil fuels will eventually run out.
2. Burning fossil fuels gives rise to gases, which cause environmental pollution and global warming.

6.2. The Thermodynamic Process

The basic function of the thermal power station is to convert energy contained in fuel to electricity. Power engineers are required to perform two-part basic calculations regarding the key parameters of a power plant. The first part is the heat energy produced by fuel combustion that is converted to electrical energy, which can be calculated by knowing the efficiency of the boiler and combustion. The efficiency can be about 80 – 90% on an HHV basis, which is the normal range for a well-optimized power plant. Second part is the steam cycle efficiency. Modern Rankine cycle, adopted in coal fired power plants, have efficiencies that vary from 30 – 40%, depending mainly on the steam parameters.

6.2.1. Heat Energy

The first parameter required to know is how much energy is contained in the fuel, the quantity and its cost. The heat energy equivalent of 1kWh of electrical energy is 860 kcal or 3412.14 Btu, which can then be expressed as heat equivalent of electrical energy and is 860 kcal/kWh or 3412.14 Btu/kWh. The quantity of fuel (m_F) can be determined by knowing the energy content of coal in terms of kilocalories per kilogram of coal as the calorific value (CV) or the higher heating value (HHV) of coal, which vary, depending on the quality and type of the coal. The heat rate is calculated from the thermal efficiency of the system. Heat rate is the heat input required to produce one unit of electricity (1kWh). One kW is 3600 kJ/hr. If the energy conversion is 100% efficient, then to produce one unit of electricity we require 3600 kJ equivalent of heat energy (or 860 kcal). The amount of fuel required can be calculated by knowing the quantity of heat and the calorific value of fuel. Thus if Q is the quantity of heat input to the system or the heat rate, then the quantity of fuel consumed is:

$$m_F = \frac{Q}{CV} \qquad 6.1$$

The heat energy output Q_0 can then be calculated by knowing the heat rate Q and the efficiency η of the system:

$$\eta = \frac{Q_0}{Q} \qquad 6.2$$

Since the energy output of a thermal power station is in terms of kWh, the heat energy output is converted to equivalent electrical energy.

Example 6.1: At atmospheric pressure (zero bar gauge or absolute 101.33 kN/m^2), water boils at 100^{0}C. 419 kJ of energy is required to heat 1 kg of water from 0^{0}C to the saturation temperature of 100^{0}C. Calculate the energy per kg available in water at this temperature.

At zero bar gauge (absolute 101.33 kN/m^2) and 100^{0}C, the specific enthalpy of water is 419 kJ/kg. Another 2257 kJ (latent heat of steam or heat of vaporization of water) of energy is required to evaporate the 1 kg of water at 100^{0}C to steam at 100^{0}C. Therefore, at zero bar gauge (absolute 101.33 kN/m^2)

the specific enthalpy of evaporation is 2257 kJ/kg. The total specific enthalpy of the steam (or heat required to evaporate water to steam) at atmospheric pressure and 100^0C can be summarized as:

$$Q_S = 419 + 2257 = 2676 \text{ kJ/kg}$$

The energy E in kWh available per kg of steam is then:

$$E = 2676 \text{ (kJ/kg)} / 3600 = \mathbf{0.74\ kWh/kg}$$

Example 6.2: A 100 MW thermal power station uses coal with calorific value of 7000 kcal/kg as fuel. If the overall efficiency of the station is 30% and the station is delivering 75% of the rated power, determine the coal consumption per hour.

Given that:

$$P = 100 \text{ MW or } 100 \times 10^3 \text{ kW}$$

$$\eta = 30\% \text{ or } 0.3$$

Electrical energy generated per hour at 75% of the rated power is:

$$E = 0.75 \times 100 \times 10^3 \times 1 = 75 \times 10^3 \text{ kWh}$$

The equivalent heat energy of 75×10^3 is:

$$Q_0 = 75 \times 10^3 \times 860 = 645 \times 10^5 \text{ kcal}$$

Knowing the overall efficiency, the heat input Q to the system is:

$$\eta = \frac{Q_0}{Q}$$

Or

$$Q = \frac{Q_0}{\eta} = \frac{645 \times 10^5}{0.3} = 215 \times 10^6 \text{ kcal}$$

The quantity of fuel m_F is then determined as:

$$m_F = \frac{Q}{CV} = \frac{215 \times 10^6}{7000} = \mathbf{30714.28\ kg}$$

Example 6.3: A 70 MW coal fired thermal power station uses coal of calorific value 7600 kcal / kg, which is consumed at a rate of 0.55 kg per kWh. If the station is working on a load factor of 60% on daily basis, calculate the efficiency and coal consumption per day.

Given that:

Calorific value of fuel = CV = 7,600 kcal/kg

Fuel consumption per kWh = m_F = 0.55 kg

Load factor = F_{LD} = 60% or 0.60

Maximum demand can be taken as the capacity of the power station, which is 70MW. Therefore the average energy delivered by the power station is:

$$E = F_{LD} \times P_m \times 24 = 0.60 \times 70 \times 10^3 \times 24 = 1.008 \times 10^6 \text{ kWh}$$

For efficiency, we use:

$$\eta = \frac{Q_0}{Q}$$

Where: $Q = CV \times m_F$

The quantity of fuel consumed daily or coal consumption per day is therefore:

$$m_F = 1.008 \times 10^6 \times 0.55 = \mathbf{554400\ kg}$$

Therefore: $Q = 7600 \times 554400 = 4213.44 \times 10^6$ kcal

The station output is 1.008 x 10^6 kWh daily, therefore the equivalent heat output Q_0 is:

$$Q_0 = 860 \times 1.008 \times 10^6 = 866.88 \times 10^6 \text{ kcal}$$

Thus using: $\eta = \frac{Q_0}{Q} \times 100 = \frac{866.88 \times 10^6}{4213.44 \times 10^6} \times 100 = \mathbf{20.57\%}$

6.2.2. The Rankine Cycle

Described in 1859 by William Rankine, it is used as a standard for understanding the performance of steam power plants. The Rankine cycle is the fundamental operating cycle of all power plants where an operating fluid is continuously evaporated and condensed. The selection of operating fluid depends mainly on the available temperature range. Figure 6.2 shows the idealized Rankine cycle system. The major components of a Rankine cycle system are; pump, boiler, turbine and condenser. In the Rankine cycle, the working substance of the engine undergoes four successive changes:

1. Heating at constant volume (as in a boiler).
2. Evaporation and superheating (if any) at constant pressure.
3. Expansion in the engine or turbine to do useful work.
4. Condensation at constant pressure with return of the fluid to the boiler.

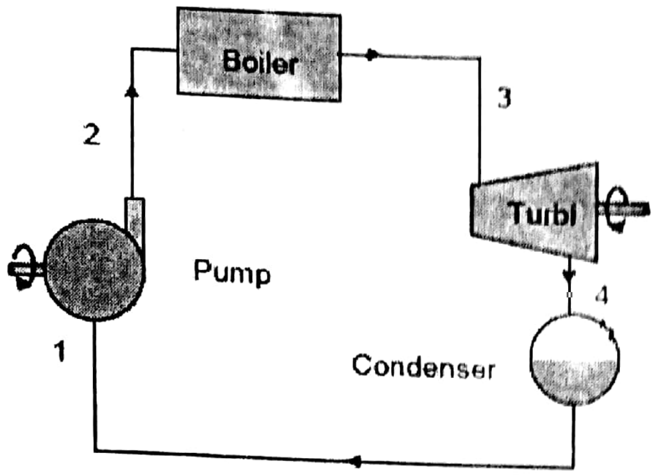


Figure 6.2: The idealized Rankine Cycle System

The operation of a system using Rankine cycle can be understood by considering the temperature-entropy (T-s) diagram shown in figure 6.3, which illustrates that the Rankine cycle operates according to the following steps:

1-2: Isentropic compression in pump (work input).
2-3: Isobaric heat transfer (constant pressure heat addition in boiler).
3-4: Isentropic expansion in turbine (work output).
4-1: Isobaric heat rejection (constant pressure heat rejection in a condenser).

The process taking place according to the steps mentioned are discussed with reference to figure 6.3 as follows:

1-2 Isentropic Compression: High pressure fluid (water) enters the boiler from the feed pump and is heated to the saturation temperature. Further addition of heat energy causes evaporation of water until it is fully converted to saturated vapour or steam. As shown in the T-s diagram of figure 6.3, water enters the pump at state 1 as saturated liquid and is compressed isentropically to the operating pressure of the boiler. In isentropic process a change takes place without any increase or decrease in entropy (constant entropy), such a process is both reversible and adiabatic. The water temperature increases somewhat during this isentropic compression process due to slight decrease in the specific volume of the water in state 2. The vertical distance between state 1 and 2 on the T-s diagram is greatly exaggerated for clarity.

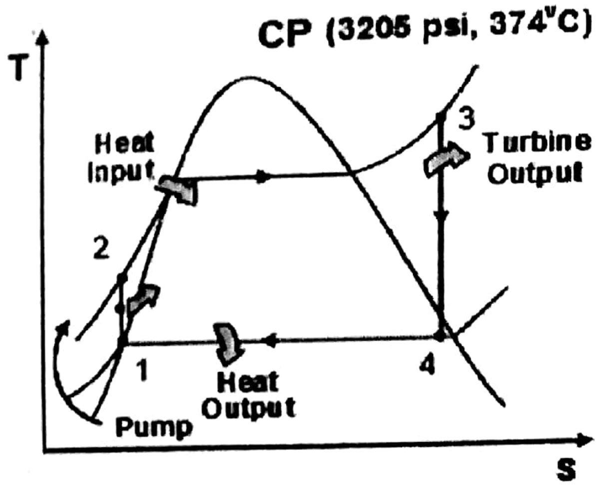


Figure 6.3: Rankine Cycle, T-s Diagram

2–3 Isobaric Heat Transfer: An isobaric process is a thermodynamic process in which the pressure remains constant. In this process the volume is allowed to increase or decrease in such a way so as to neutralize any pressure changes that would be caused by heat transfer. Water enters the boiler as a compressed liquid at state 2 and leaves as a superheated vapor at state 3 as shown in figure 6.3. The boiler is basically a large heat exchanger where the heat originating from combustion gases, is transferred to the water essentially at constant pressure. The boiler together with the section where the steam is superheated (the superheater) is often called the steam generator.

3-4 Isentropic Expansion: The superheated vapor (steam) at state 3 enters the turbine, where it expands isentropically. The vapor or steam is expanded in the turbine, thus doing work which may be converted to electrical energy when an electric generator is coupled to the turbine shaft. In practice, the expansion is limited by the temperature of the cooling medium and by the erosion of the turbine blades by fluid in the vapor stream as the process moves further into the two-phase region (vapor and liquid). Exit vapor qualities should be greater than 90%. The pressure and the temperature of steam drops during this process to the values at state 4; where the steam enters the condenser.

4-1 Isobaric Heat Rejection: The vapor-fluid mixture leaving the turbine at 4 as shown in the T-s diagram of figure 6.3 is condensed at low pressure, usually in a condenser, which is basically a large heat exchanger with circulating cooling water taken from a source such as a lake or a river. Steam is condensed at constant pressure by rejecting heat to the cooling water. In well designed and maintained condensers, the pressure of the vapor is well below atmospheric pressure, approaching the saturation pressure of the operating fluid at the cooling water temperature. The pressure of the condensate is raised in the feed pump. Because of the low specific volume of liquid fluids, the pump work is relatively small and often neglected in thermodynamic calculations.

The components of the Rankine cycle; pump, boiler, turbine and condenser are steady-flow devices. All four processes of the Rankine cycle can therefore be analyzed as steady-flow process. Values of heat and work can be determined by applying the first law of thermodynamics to each step. When it comes to the design of a power plant, the first law of thermodynamics will be much more useful if we can express it as:

$$\Delta H = Q + W \qquad 6.3$$

In equation 6.3, ΔH is the change in internal energy of the system or enthalpy, Q is the heat transferred into/out of the system and W is the work done by/on the system. This reformulation of the first law of thermodynamics tells us that when we define a system the energy of the system will remain constant unless there is heat added or taken away from the system, or some work is done. According to the law of thermodynamics, the steady flow equations of the process for per unit mass are:

Pump ($Q = 0$): $W_{in} = (H_2 - H_1)$

Boiler ($W = 0$): $Q_{in} = (H_3 - H_2) \approx (H_3 - H_1)$ because $H_1 \approx H_2$

Turbine ($Q = 0$): $W_{out} = (H_4 - H_3)$

Condenser ($W = 0$): $Q_{out} = (H_1 - H_4)$

Or $$(Q_{in} - Q_{out}) + (W_{in} - W_{out}) = 2(H_3 - H_1)$$

Or $$(Q_{in} - Q_{out}) + (W_{in} - W_{out}) = \Delta H \qquad 6.4$$

W represents the work done and Q represents the heat value, then the efficiency of cycle, according to laws of thermodynamics is defined as:

$$\eta = \frac{W_{net}}{Q_{in}} \qquad 6.5$$

$$\eta = \frac{-(H_4 - H_3) + (H_2 - H_1)}{(H_3 - H_2)}$$

Since $(H_2 - H_1)$ is neglected, then:

$$\eta = \frac{(H_3 - H_4)}{(H_3 - H_2)} \qquad 6.6$$

The work done by the pump can be obtained in terms of pressure. If the working fluid is assumed to be incompressible (constant volume V), the increase in enthalpy for isentropic compression is then given by:

$$dQ = dH - Vdp = 0$$

Therefore: $$dH = Vdp$$

Or $$\int_1^2 dH = V\int_1^2 dp$$

Or $$W_{pump}(in) = (H_2 - H_1) = V(p_2 - p_1) \qquad 6.7$$

The enthalpy for a particular pressure and temperature can be obtained from the heat-entropy chart (Mollier diagram) or from steam tables given in Appendix B or from standard steam tables given in most books on engineering thermodynamics. However, in real plants each stage of the Rankine cycle is associated with irreversible processes, thus reducing the overall efficiency. The actual vapor power cycle differs from the ideal Rankine cycle as a result of irreversibilites in various components. The situation is illustrated in figure 6.4.

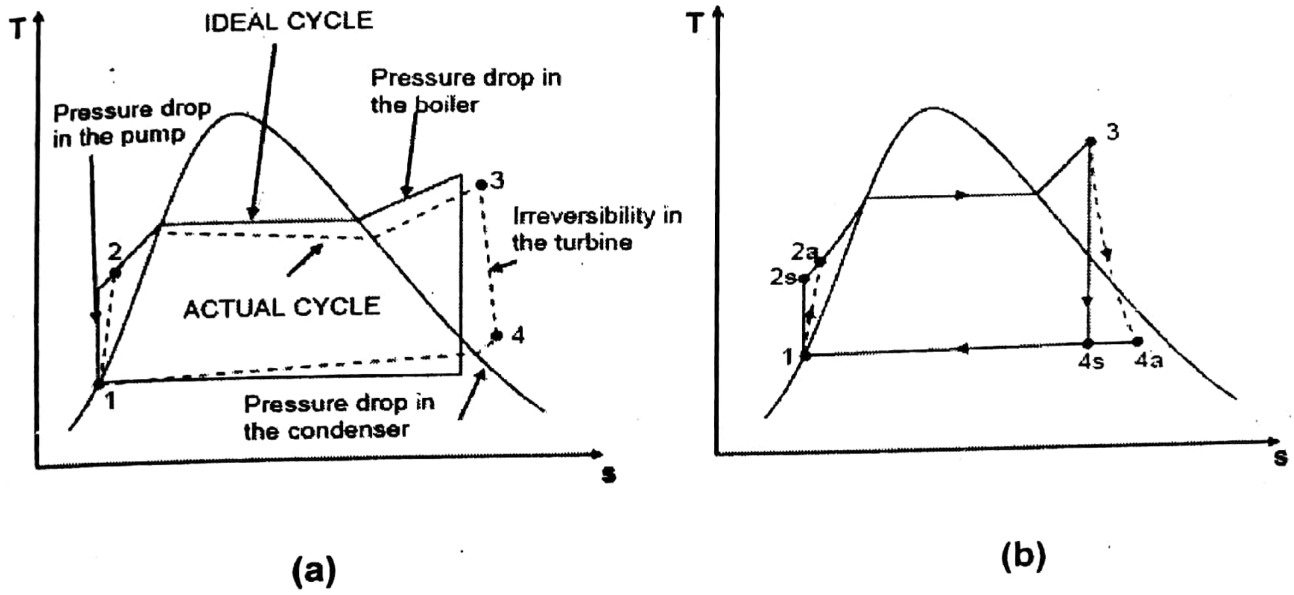


Figure 6.4: Deviation from Ideal Rankine Cycle

Fluid friction and heat loss to the surroundings are the two common sources of irreversibilites. The major source of irreversibility is the heat loss from the steam to the surrounding as the steam flows through various components in the thermal power plant. Fluid friction causes pressure drop in the boiler, the condenser and the piping between various components. Of particular importance are the irreversibilites occurring within the pump and the turbine. The pressure at the turbine inlet is somewhat lower than that at the boiler exit due to the pressure drop in the connecting pipes. As a result of irreversibilities a pump requires a greater work input, and a turbine produces a smaller work output. Under ideal condition the flow through these devices is isentropic. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle. This requires a large pump and larger work input to the pump, the situation is shown in figure 6.4(b). Turbine and pump irreversibility can be included in the calculation of the overall cycle efficiency by defining mechanical efficiency of the turbine η_m.

$$\eta_m = \frac{W_a}{W_S} = \frac{(Q_{in} - Q_{out})_a}{(Q_{in} - Q_{out})_S} \qquad 6.8$$

Where subscript 'a' indicates actual values and subscript 's' indicates isentropic values. If the pump efficiency is η_p, then:

$$\eta_p = \frac{W_S}{W_a} = \frac{V(p_2 - p_1)}{W_a} \qquad 6.9$$

If η_m and η_p are known, the actual enthalpy after the compression and expansion steps can be determined from the values for the isentropic processes. The turbine inefficiency directly reduces the work produced in the turbine and, therefore the overall efficiency. The inefficiency of the pump increases the enthalpy of the liquid leaving the pump and, therefore, reduces the amount of energy required to evaporate the liquid. However, the energy to drive the pump is usually more expensive than the energy to feed the boiler. The result is that the efficiency of the Rankine cycle is less than that of the ideal vapor cycle.

Example 6.4: Determine the work done and efficiency of the ideal Rankine cycle for boiler pressure of 25 bars and condenser pressure of 0.15 bars.

Given that:

p_b = 25 bars

p_C = 0.15 bars

Where p_b and p_C are the boiler and condenser pressure respectively. Refer to figure 6.3. From steam tables or heat-entropy chart, the total entropy of dry saturated steam at pressure of 25 bars is 6.254 kJ/kg ^{0}K and enthalpy is 2801 kJ/kg (assuming dryness fraction of unity). Thus:

$H_3 = 2801$ kJ/kg

$s_3 = 6.254$ kJ/kg ^{0}K

In case of ideal expamsion, $s_3 = s_4$.

Then: $s_4 = s_f + x_4 s_{fg}$

The quantities s_f and s_{fg} are obtained from the steam tables of heat-entropy chart for pressure 0.15 bars at the turbine outlet.

Or $$x_4 = \frac{6.254 - 0.755}{7.254} = 0.758$$

The enthalpy of steam at the output of the turbine or at the condenser at pressure of 0.15 bars with the help of steam tables or heat-entropy chart is determined as follows:

$$H_4 = h_f + x_4 h_{fg}$$

The values of h_f and h_{fg} are obtained from standard steam tables for pressure of 0.15 bars at the condenser.

Therefore: $H_4 = 226 + 0.758(2373.2) = 2024.88$ kJ/kg

From steam tables, the enthalpy of water conversion from steam in the condenser at the pump or boiler input is 226 kJ/kg for pressure of 0.15 bars. Therefore:

$$H_1 = 226 \text{ kJ/kg}$$

Therefore the heat input from the boiler to the turbine input is:

$$Q = H_3 - H_1 = 2801 - 226 = 2575 \text{ kJ/kg}$$

The work done is the output of the turbine, which is:

$$P = H_3 - H_4 = 2801 - 2024.88 = \mathbf{776.12\ kJ/kg}$$

The efficiency of the Rankine system is therefore:

$$\eta = \frac{P}{Q} \times 100 = \frac{776.12}{2575} = \mathbf{30.14\%}$$

It must be noted that Rankine cycle has a low efficiency. The efficiency of even the ideal Rankine cycle, with an isentropic turbine and pump drops below the efficiency of the Carnot cycle. The followings are the main factors, which tends to reduce the overall efficiency of the thermal plant:

1. Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. The feed water

is always a sub-cooled liquid and the effluent is sometimes a saturated vapor, but more often it is a superheated vapor. Therefore, lowering the operating pressure of the condenser automatically lower the temperature of the steam, and thus the temperature at which heat is rejected. The effect of lowering the condenser pressure on the Rankine cycle efficiency is illustrated in figure 6.5.

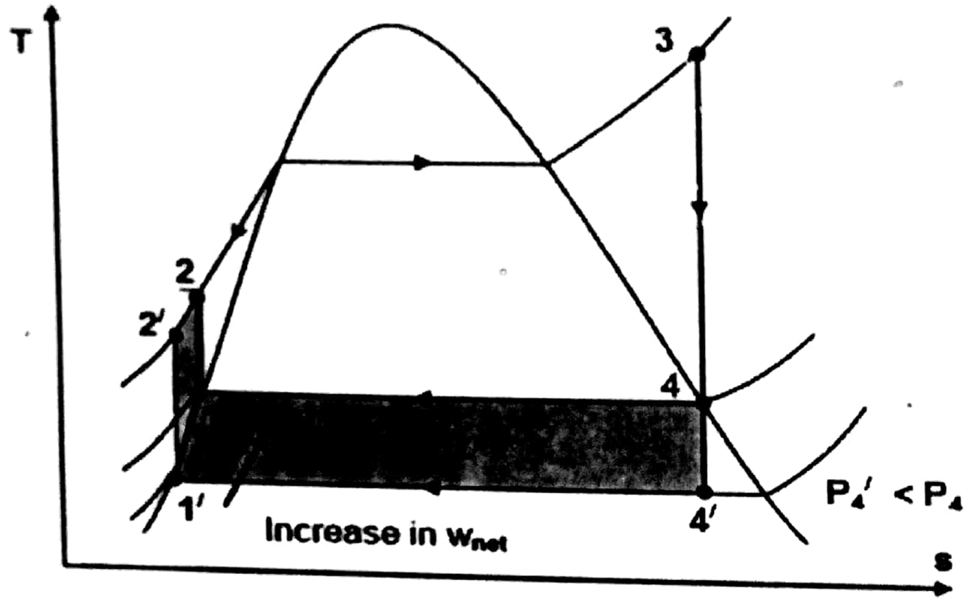


Figure 6.5: Effect of Lowering Condenser Pressure

However, the disadvantage of lowering the condenser pressure is increase in the moisture content of the steam at the final stages of the turbine and therefore the work done by the turbine will be less and tend to decrease the turbine efficiency. The presence of large quantities of moisture is also highly undesirable in turbines because it erodes the turbine blades. This new source of irreversibility is a small price to pay for saving the pump and turbine from costly damage.

2. The change from the Carnot cycle to the Rankine cycle is in fact that the boiler component is not isothermal, which makes it very difficult to transfer heat to the working fluid in a reversible manner. Even the most sophisticated boilers transform only 40% of the fuel energy into useable steam energy. There are two main reasons for this wastage; firstly, the combustion gas temperatures are

between 1000°C and 2000°C, which is considerably higher than the highest vapor temperatures and the transfer of heat across a large temperature difference, increases the entropy. Secondly, combustion (oxidation) at technically feasible temperatures is highly irreversible. The average temperature at which heat is added to the steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures, which has very desirable effect as can be seen in T-*s* diagram as shown in figure 6.6. It decreases the moisture content of the steam at the turbine exit

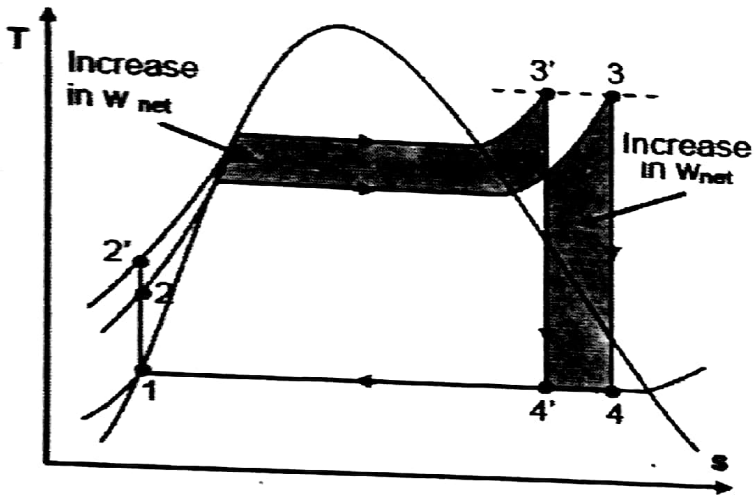


Figure 6.6: Rankine Cycle with Vapor Superheating

As mentioned earlier, the temperature to which steam can be superheated is, however, also limited by the metallurgical consideration.

3. The average temperature during the heat addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which boiling take place. This in turn raises the average temperature at which heat is added to the steam and thus raises the thermal efficiency of the cycle. The isentropic turbine in a Rankine cycle takes in saturated or superheated vapor and exhausts a high-quality vapor liquid mixture. The quality of the turbine effluent is usually greater than 90%. The condenser in the Rankine cycle produces a saturated liquid. Since the heat transfer surface in the condenser has a finite value, the condensation will occur at a temperature higher than the

temperature of the cooling medium. Again, heat transfer occurs across a temperature difference, causing the generation of entropy. Moreover, the deposition of dirt in condensers during operation with cooling water reduces the efficiency. The isentropic pump raises the pressure of the saturated liquid feed, producing a sub-cooled liquid thus completing the cycle.

Example 6.5: Calculate the ideal and actual steam consumption of a steam engine working with dry steam from the boiler in example 6.4 between the given pressures if the engine efficiency is 62%.

The efficiency of the Rankine cycle in example 6.4 between the given pressures is calculated to be 30.14%. The efficiency of the steam engine is 62%. The work done as calculated in example 6.4 is:

= 776.12 kJ/kg

The ideal steam consumption is:

$$=\frac{0.746\times3600}{776.12}= \textbf{3.46 kg per HP hour}$$

The actual steam consumption is:

$$=\frac{3.46}{0.62}= \textbf{5.58 kg per HP hour}$$

Example 6.6: A 30MW steam turbine in a thermal power station takes superheated steam at 70 bars and at 500°C. The steam expands adiabatically with an isentropic efficiency of 88% to a pressure of 0.1 bars. Calculate the enthalpy at the turbine output and the required rate of steam flow in kg/s.

Given that:

Power output P = 30MW = 30000 kW

Isentropic efficiency η = 88% or 0.88

Refer to figure 6.3. From the thermodynamic steam tables; at 500°C and 70 bars, the enthaply of the superheated steam at the turbine is:

$H_3 = 3410.6$ kJ/kg

From steam tables, the corresponding value of entropy is:

$s_3 = 6.799$kJ/kg °K

For an ideal expansion: $s_3 = s_4$. Then: $s_4 = s_f + x_4 s_{fg}$. Where x_4 is the dryness fraction (steam quality). Using steam tables:

$$6.799 = 0.649 + x_4(7.502)$$

From which: $x_4 = 0.8197$ kJ/kg ^{0}K. The enthalpy of steam after exiting the turbine at 0.1 bar is then:

$$H_4 = h_f + x_4 h_{fg}$$

Using steam tables, we have:

$$H_4 = 191.8 + 0.8197 \times 2392.9 = 2153.26 \text{ kJ/kg}$$

Ideal change in the enthalpy:

$$\Delta H' = H_4' - H_3 = 2153.26 - 3410.6 = -1257.34 \text{ kJ/kg}$$

Actual change in enthalpy is therefore:

$$\Delta H = \eta \times \Delta H' = 0.88 \times (-1257.34) = -1106.46 \text{ kJ/kg}$$

Also: $\Delta H = H_3 - H_4 = -1106.46$ kJ/kg

From which: $H_4 = -1106.46 + 3410.6$ = **2304.14 kJ/kg**

Using the law of thermodynamics:

$$Q + W = \Delta H$$

At the turbine Q = 0, therefore:

$W = \Delta H \Rightarrow P = \dfrac{\Delta H}{t}$: by dividing both sides by time t. Then the work done per unit time is the power P.

or $-P = m\Delta H$

The negative sign is added to P since this is the output power from the turbine.

Therefore: $m = \dfrac{-P}{\Delta H} = \dfrac{-30000}{-1106.46}$ = **27.11 kg/sec**

Example 6.7: A Rankine cycle works between 40 bars and 400^{0}C at the boiler exit and 0.035 bars at the condenser. Assuming isentropic expansion, calculate the efficiency.

Refer to figure 6.3, steam is superheated at 400^{0}C and 40 bars at the output of the boiler. The steam enters the turbine system, where it expands thus doing work and the exit pressure from the turbine at the input of condenser is

0.035 bars. From the steam tables, at boiler exit temperature and pressure, the enthalpy of the steam is:

$$H_3 = 3215.7 \text{ kJ/kg}$$

From thermodynamic tables of superheated steam, the entropy of steam at 400°C and at 40 bars is:

$$s_3 = 6.773 \text{kJ/kg } ^0\text{K} \text{ (from superheated steam table)}$$

For ideal expansion: $s_3 = s_4$. Then:

$$s_4 = s_f + x_4 s_{fg}$$

At the condenser where the pressure is 0.035 bars, from the steam tables the corresponding values for the above expression are substituted to calculate the dryness fraction x_4, we have:

or $$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.773 - 0.391}{8.133} = 0.784$$

Therefore: $$H_4 = h_f + x_4 h_{fg} = 111.8 + 0.784 \times 2438.6 = 2023.66 \text{ kJ/kg}$$

At the pump H_1 at 0.035 bars is 111.8 kJ/kg. Therefore the heat input Q to the turbine from the boiler exit is:

$$Q = H_3 - H_1 = 3215.7 - 111.8 = 3103.9 \text{ kJ/kg}$$

The power output P of the turbine is the work done by the expansion of steam from boiler pressure to the condenser pressure:

$$P = H_3 - H_4 = 3215.7 - 2023.66 = 1192.04 \text{ kJ/kg}$$

The efficiency is therefore:

$$\eta = \frac{P}{Q} = \frac{1192.04}{3103.9} = 0.384 \text{ or } \mathbf{38.4\%}$$

The irreversibility of any process is reduced if it is performed as close as possible to the temperatures of the high temperature and low temperature reservoirs. This is achieved by operating the condenser at sub-atmospheric pressure. The temperature in the boiler is, however, limited by the saturation pressure. In most thermal power stations of large capacity, preheating feed-water and turbine stages are used to increase the efficiency as shown in figure 6.7.

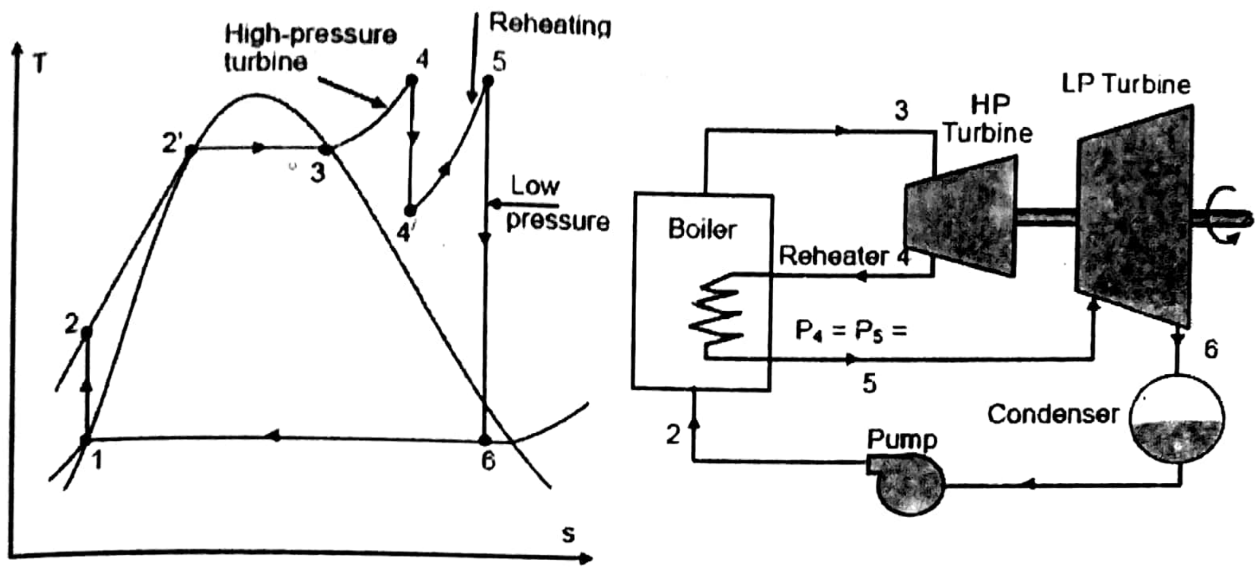


Figure 6.7: (a) T-s Diagram (b) Schematic Diagram

To preheat the feed-water to its saturation temperature, bleeding vapor from various positions of the turbine which is then passed through external heat exchangers (regenerators) is used as shown in figure 6.8.

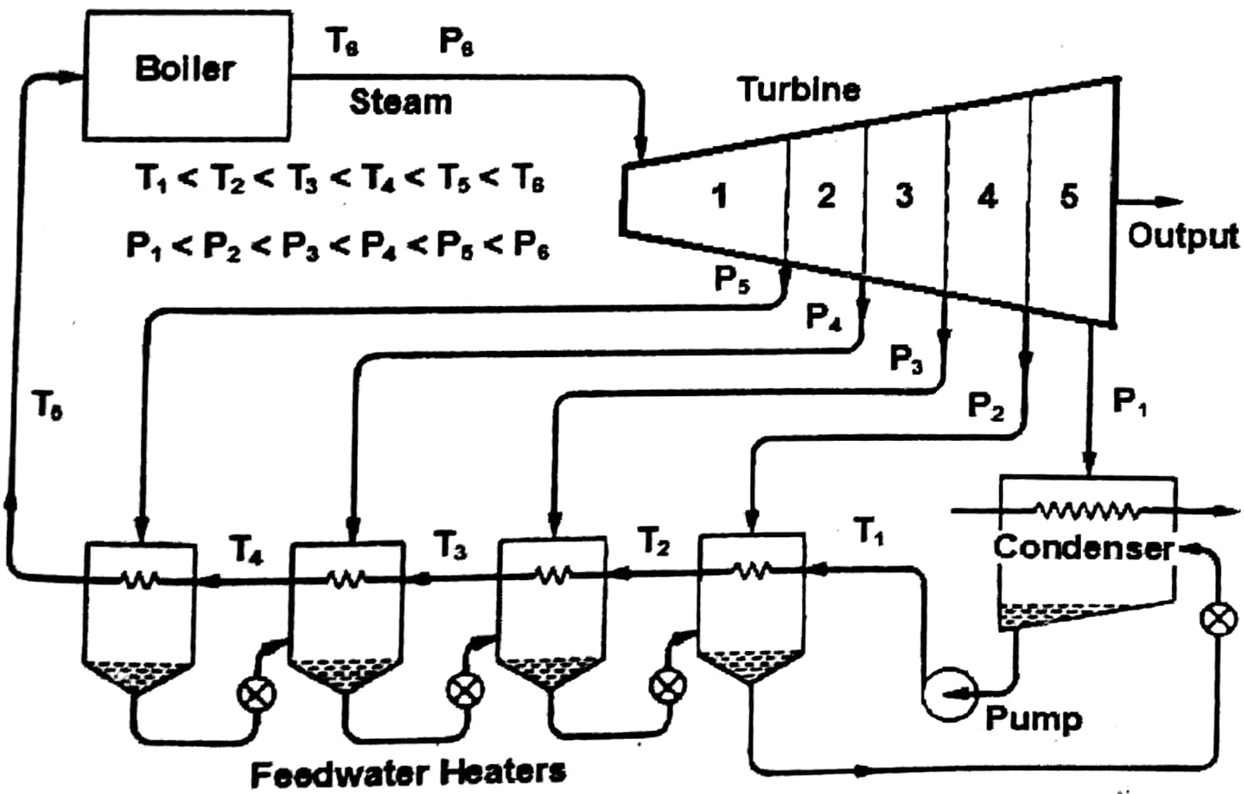


Figure 6.8: Regenerative Feed-Water Heating System

It is quite common to reheat the vapor after expansion in the high pressure turbine and expand the reheated vapor in a second, low pressure turbine. The resulting irreversibility reduces the efficiency of the boiler. According to the Carnot thermodynamic cycle, the highest efficiency is reached if heat transfer occurs isothermally. The cold liquid leaving the feed pump is mixed with the saturated liquid in the boiler and/or re-heated to the boiling temperature.

Ideally, the temperature of the bleed steam should be as close as possible to the temperature of the feed liquid. The average temperature at which the heat is supplied for the cycle with a higher maximum pressure is increased over the original cycle, so that the efficiency increases. The T-s diagram of the situation is shown in figure 6.9. The steam is bled from the turbine system to heat the feed water. Knowing the bleed steam temperature and pressure the efficiency of the system can be determined. The high combustion temperature of the fuel is better utilized if a gas turbine or Brayton engine is used as 'topping cycle' in conjunction with a Rankine cycle, referred to as combined cycle, which is discussed later in this book. In this case, the hot gas leaving the turbine is used to provide the energy input to the boiler.

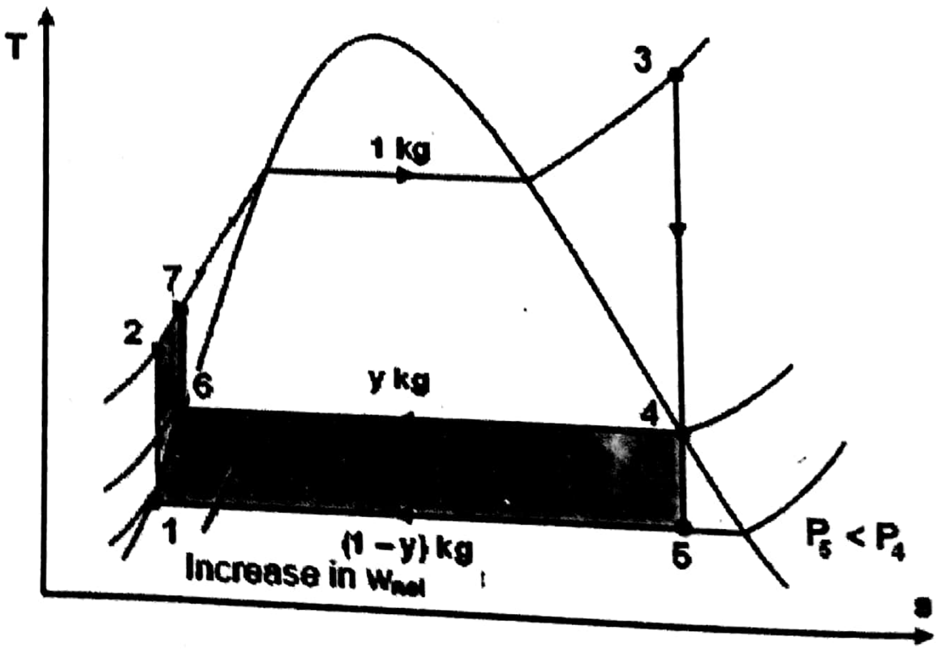


Figure 6.9: T-s Diagram

In co-generation systems, the energy rejected by the Rankine cycle is used for space heating, process steam or other low temperature applications.

Example 6.8: Calculate the efficiency when a feed water heater is used with the Rankine cycle system described in example 6.7.

The pressure and temperature of the bleed steam can be determined from steam tables: At 40 bars the steam temperature is 250.3°C and at 0.035 bars pressure the temperature is 26.69°C. Thus the steam bleed temperature for one feed heater will be the average temperature between the two pressures:

Therefore:

$$t_S = \frac{26.69 + 250.3}{2} = 138.5^0 C$$

This temperature of 138.5°C corresponds to 3.5 bars pressure acording to steam tables. Refer to the figure 6.9. Thus steam is bled at pressure of 3.5 bars in the turbine to heat the feedwater to the boiler. Thus from the figure, the path 6-7-3-4-6 is followed. From example 6.7:

$$H_3 = 3215.7 \text{ kJ/kg}$$

The corresponding entropy of superheated steam at 40 bars pressure is: $s_3 = 6.773$kJ/kg °K. For ideal expansion: $s_3 = s_4$. Thus:

$$s_4 = s_f + x_4 s_{fg}$$

or

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.773 - 1.727}{5.212} = 0.968$$

The enthalpy at 3.5 bars pressure is calculated as:

$$H_4 = h_f + x_4 h_{fg} = 584.3 + 0.968 \times 2147.3 = 2662.88 \text{kJ/kg}$$

And at 0.035 bars pressure the enthalpy at the pump of boiler input, from the steam tables is:

$$H_6 = H_7 = h_f = 584.3 \text{ kJ/kg}$$

The enthalpy of steam H_5 after it has expanded in the turbine to the condenser pressure is calculated as: For ideal expansion: $s_4 = s_5$. Thus:

$$s_5 = s_f + x_5 s_{fg}$$

or $$x_5 = \frac{s_5 - s_f}{s_{fg}} = \frac{6.773 - 0.391}{8.133} = 0.784$$

The enthalpy at 0.035 bars pressure at the turbine output is calculated as:

$$H_5 = h_f + x_5 h_{fg} = 111.8 + 0.784 \times 2438.6 = 2023.66 \text{ kJ/kg}$$

At pressure of 0.035 bars, the enthalpy at the pump or boiler input is:

$$H_1 = H_2 = h_f = 111.8 \text{ kJ/kg (from steam tables)}$$

Heat input to the boiler is therefore:

$$Q = H_3 - H_7 = 3215.7 - 584.3 = 2631.4 \text{ kJ/kg}$$

From figure 6.9, the energy balance equation for heat transfer is:

$$yH_4 + (1-y)H_2 = H_6$$

or $$y(2662.88) + (1-y)111.8 = 584.3$$

From which: y = 0.185. Thus the heat output from the turbine is:

$$Q_0 = (1-y)(H_5 - H_1)$$

$$Q_0 = (1-0.185)(2023.66-111.8) = 1558.16 \text{ kJ/kg}$$

The power output from the turbine is:

$$P = Q - Q_0 = 2631.4 - 1558.16 = 1073.24 \text{ kJ/kg}$$

The efficiency is therefore:

$$\eta = \frac{P}{Q} = \frac{1073.24}{2631.4} = 0.4078 \text{ or } \mathbf{40.78\%}$$

There is an improvement of 40.78 – 38.4 = **2.38%** by incorporating a single feedwater heater for which steam is bleed from the turbine at a pressure of 3.5 bars to heat the incoming water to the boiler.

Example 6.9: By how much percent the efficiency will increase by superheating the steam at 300^0C in example 6.4.

From steam tables or heat entropy chart, at 25 bars and at 300^0C, the enthalpy of superheated steam is:

$$H_3 = 3010 \text{ kJ/kg}$$

Using steam tables or heat-entropy chart, the entropy at at 25 bars pressure and at 300^{0}C temperature is:

$$s_3 = 6.647 \text{ kJ/kg } {}^0\text{K}$$

For ideal expansion: $s_3 = s_4$, therefore:

$$s_4 = s_f + x_4 s_{fg}$$

or
$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.647 - 0.755}{7.254} = 0.8122$$

The enthalpy at 0.15 bars pressure at the turbine output is calculated as:

$$H_4 = h_f + x_4 h_{fg} = 226 + 0.8122 \times 2373.2 = 2153.51 \text{kJ/kg}$$

And at 0.15 bars pressure the enthalpy at the pump of boiler input, from the steam tables is:

$$H_1 = h_f = 226 \text{ kJ/kg}$$

At the pump H_1 at 0.15 bars is 226 kJ/kg. Therefore the heat input Q to the turbine from the boiler exit is:

$$Q = H_3 - H_1 = 3010.4 - 226 = 2784.4 \text{ kJ/kg}$$

The power output P of the turbine is the work done by the expansion of steam from boiler pressure to the condenser pressure:

$$P = H_3 - H_4 = 3010.4 - 2153.51 = 856.89 \text{ kJ/kg}$$

The efficiency is therefore:

$$\eta = \frac{P}{Q} = \frac{856.89}{2784.4} = 0.3077 \text{ or } \mathbf{30.77\%}$$

Thus there is increase of 30.77 – 30.14 = **0.66%** when the steam is superheated to 300^{0}C at the same pressure of 25 bars. The efficiency can be increased by either superheating at higher pressure at the same temperature of 300^{0}C or superheating at higher temperature at the same pressure of 25 bars.

6.3. Thermodynamic Circuits of Thermal Power Plant

There are four main circuits which would give an idea about the working of different components of a thermal power plant. These are discussed as follows:

6.3.1. Coal-Ash Circuit

This circuit deals mainly with feeding the boiler's furnace with coal for combustion purposes and taking care of the ash that is generated during the combustion process and includes equipment and accessories that are used to handle the transfer and storage of coal and ash and is shown in figure 6.10. The major components of this circuit are coal mill, burner, furnace and ash pit. The coal from the coal yard is crushed to convenient size and is conveyed and stored in the hoppers above the boiler. The coal is then passed through to the coal feeders for regulating and measuring coal quantity to a coal mill or pulverizers and then to pulverized coal bins. In coal mills, the coal is converted to powder by the following techniques:

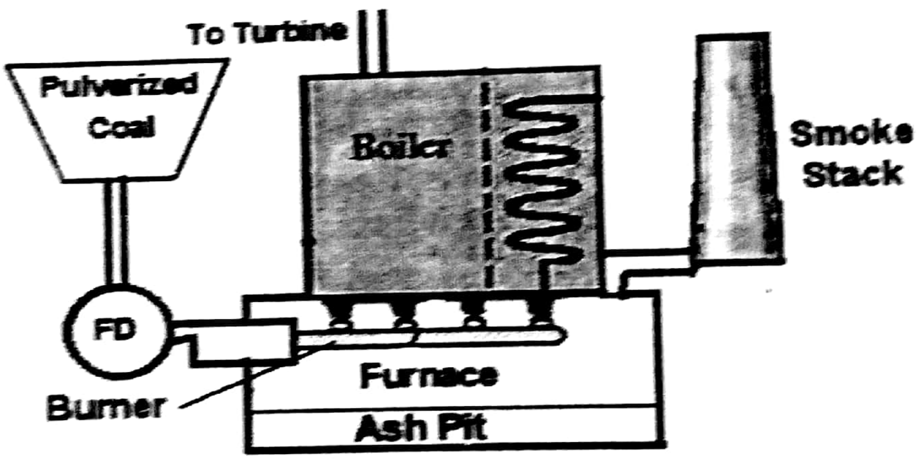
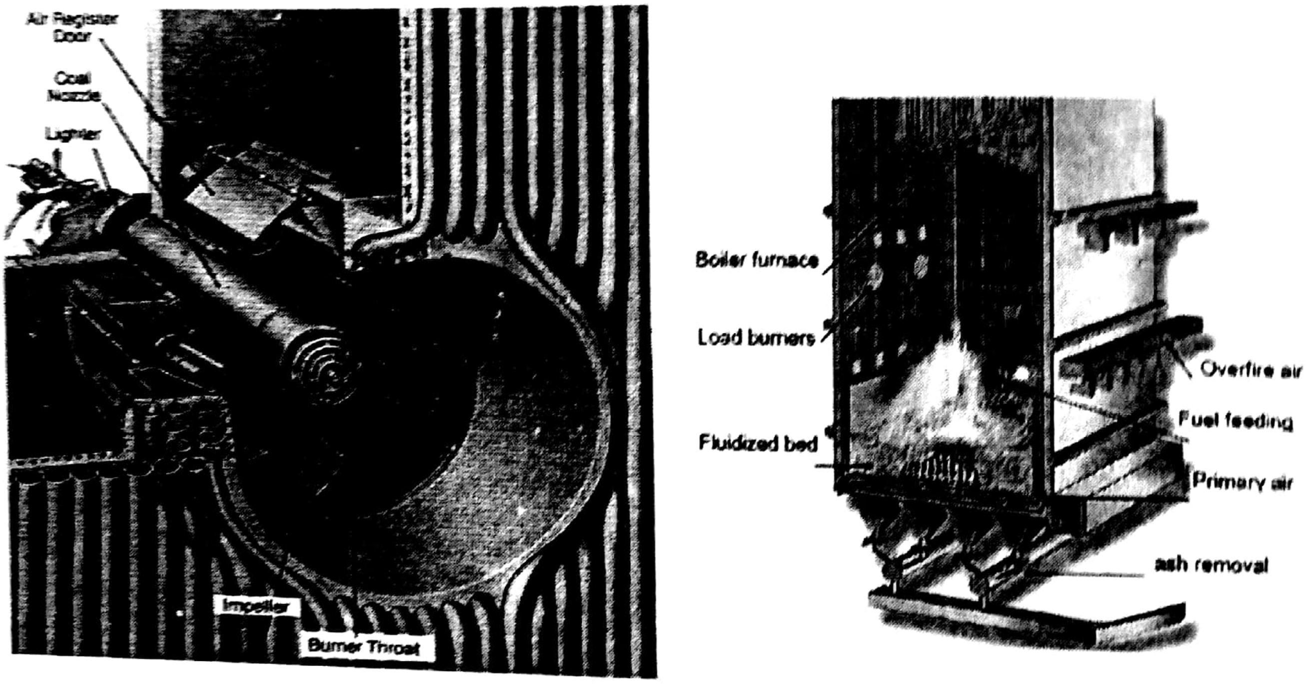


Figure 6.10: Coal and Ash Circuit

1. **Impact:** By this technique, coal is hammered thus breaking up into smaller pieces.
2. **Attrition:** In this technique, coal pieces rub against each other and metal surfaces to disintegrate into further smaller pieces.
3. **Crushing:** By this technique, the coal is crushed down to dust or powder by making it to be grind between metal rollers.

The advantage of the bin system include the ability to grind coal at a steady rate regardless of boiler load and therefore less capacity pulverizer or mill needs to be installed for a given boiler capacity. Furthermore, in such a system, the mills can be run steadily at their most efficient rating and occasional mill outage does not affect coal feed to the furnace. From the pulverized coal bin, the pulverized or powdered coal is conveyed by primary hot air injectors through coal pipes to burners of one tier or level at an angle into the furnace to give a swirling action for powdered coal for proper mixing of coal powder and also the incoming secondary hot air from forced draught (FD) fans, to give the best combustion. Primary air that carries the powdered coal from the mill to the furnace is only about 20% of the total air needed for combustion. Before the coal enters the furnace, it is mixed with secondary hot air. Too much secondary air can cool the mixture and prevent its heating to ignition temperature. The secondary air inlet is controlled by the speed of the forced draught fans. Generally, the typical air-fuel ratio of the mixture leaving the pulverizers is about 2:1. However, the air-fuel ratio is different for different types of coals and do have an impact on the combustion at the burner. In order to ensure uniform distribution of the fuel mixture supplied by any pulverizer to the burners, all fuel pipes from any particular pulverizer usually have the same length and same number of bends so as to give the same head loss between the mill and the individual burners. Division of the flow into separate pipes is either at the mill exit or at riffle boxes along the pipe. The pulverized coal is propelled by means of exhausters towards the burners, incorporated with each individual arrangement of pulverizers and burners. The individual exhauster is connected to the top of the mill by a pipe, and another pipe connects the same exhauster to burners on the furnace. The exhauster acts like a big vacuum cleaner, sucking coal powder out of the mill and then blowing it through pipes leading to the burners. The fuel mixture is conveyed in pipes to the burners, with each mill serving several burners. Figure 6.11(a) is a sectioned view of a typical burner used in thermal power plants. To provide sufficient combustion temperature in the furnace before spraying powdered coal to catch fire or ignite, the furnace temperature is brought up by spraying and burning light oil by means of igniter oil guns. Oil is used in a fine spray, as oil can catch fire even in ambient temperature.



(a)

(b)

Figure 6.11: (a) Burner (b) Furnace

Alternatively gas is also used for ignition instead of oil, if available in plenty; however, in this case the igniter gun design differs. To ignite the ignition oil or gas, an Electric High Tension (EHT) spark in the path of oil or gas is used momentarily and then the spark gun is withdrawn. Finally, the powder ignites within the furnace thus heating the water inside the boiler. If the system does not have pulverized coal bin then coal powder is conveyed directly to coal burners from pulverizes. The burners themselves are extensions of these pipes that pass through the walls of the furnace. Ignition of fuel mixture depends on the rate of flame propagation. The fuel mixture must be injected at a rate equal to the flame-front travel rate in order to ensure efficient ignition and prevent flashback into the burner. The rate depends on the velocity with which the mixture is injected. The velocity is typically 25 m/s. Too high velocity will lead to excessive erosion of the fuel pipes, particularly at the bends. Velocity limitations for both the pulverizers and the pipes mean that the primary airflow can only be reduced to about 60–

80% of the full load value. Different manufactures provide different arrangements of the burners on the furnace walls to give different flame characteristics.

Tangential Firing: In this type, the coal feeds into the furnace at the corners of the furnace. This produces a cyclone effect in the furnace thus ensuring proper mixing of air and fuel that is essential for efficient and complete combustion. The combustion air also enters into the furnace tangentially above and below these burners. Each pulverizer supplies coal to one level so that the firing is always balanced. In most of the tangential firing systems, the burners can tilt 30 degrees up or down which aids in controlling temperature and combustion conditions. These are called "tangential tilting burners."

Wall Firing: In this type of firing method, the burners are placed on the front wall or the sidewalls of the furnace. Each set of burners connects to one pulverizer to maintain balance. The air required for each burner enters the furnace adjacent to the burners.

Down-Shot Firing: In this type of firing method, the burners are placed in such a way that the flames are pointed downwards. This method is used with coals that take more time to burn, for example anthracitic coals.

Figure 6.11(b) shows a cut-away view of a typical furnace. Furnace walls are usually made up of solid refractory bricks, containing about 40% aluminum oxide, 55% silica and 5% other oxides as impurities. In some furnaces, insulating bricks with 85% magnesia, rock wool, diatomaceous earth are used, which reduces heat losses but raise the refractory temperatures. In large furnaces, steel ledges or hangers supported on steel columns or beams hold brick wall sections to provide extra load bearing strength. Fusion temperatures of refractory materials generally vary between 1450^{0}C and 2000^{0}C. In the furnace shown in figure 6.11(b), the finely powdered pulverized coal is piped into different burners placed in the furnace of the boiler, the arrangement usually used in boilers of large capacity power plants. By circulating fluidized bed where large coal particles spread on to the bottom of the furnace and the primary air pressure from the bottom keeps it floating to give good combustion. Smaller units use grate firing where the coal is spread on a moving or stationary grate, and the combustion air is admitted from below.

In some power stations residual oil is used as the main fuel. This oil congeals (becomes solid like wax) below about 50°C due to its high content of wax (about 50%). This oil therefore is always kept above this temperature even in storage tanks to make it suitable for pumping. For spraying into the furnace the oil temperature at burner tips is maintained at about 100°C. For all instruments on this oil line, lagging or heat insulation is provided for their proper working. The oil is transported directly from refinery by means of oil wagons provided with steam heating coils and is generally loaded at the refinery at about 80°C.

6.3.2. Air-Gas Circuit

Air is one of the main components of the heat producing system and thus necessary for combustion. Pre-heated air is supplied into the combustion chamber and the flue gases are exhausted to the atmosphere through air-heater, economizer, evaporato superheater and dust collectors, which forms the components of air-gas circuit. Since large amount of coal is burnt inside the furnace of boiler, it requires sufficient quantity of air which is supplied using either forced draught (FD) or induced draught (ID) fans. The exhaust gases from the combustion are used to heat the ingoing air through a heat exchanger before being allowed to escape into the atmosphere. This process is known as reheating; and is one of the techniques to increase the efficiency of the system using Rankine cycle. The main practical advantage of reheat system is the decrease in moisture content in the turbine because most of the heat addition in the cycle occurs in the vaporization part of the heat addition process. They are the same as the superheaters but their exit temperature is a little less, with pressure about 20-25% less than the superheater.

6.3.3. Feedwater-Steam Circuit

The components of this circuit are feed pump, boiler (including super-heater) and condenser as shown in figure 6.12. This circuit deals with supplying of steam generated from the boiler to the turbines and to handle the outgoing steam from the turbine by cooling it to form water in the condenser so that it can be reused in the boiler as 'make-up' water to compensate for any losses due to evaporation. Water is fed into the boiler using a feed pump. Water is evaporated inside the boiler. The vapour is fed to the superheater and the superheated steam is supplied to the turbine to generate power.

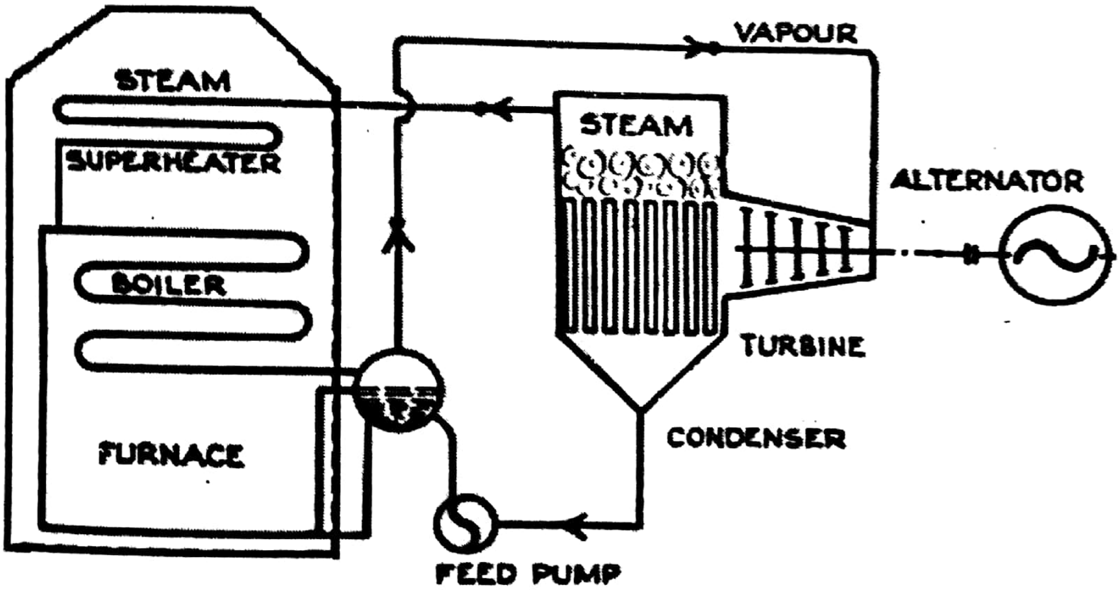


Figure 6.12: Feed-Water Circuit

6.3.4. Cooling Water Circuit

This circuit of the thermal power plant deals with handling of the cooling water required in the system and is shown schematically in figure 6.13. Since the amount of water required to cool the outgoing steam from the boiler is substantial, it is either taken from a nearby water source such as a river or lake, or it is done through evaporation in cooling towers if the quantity of cooling water available is limited.

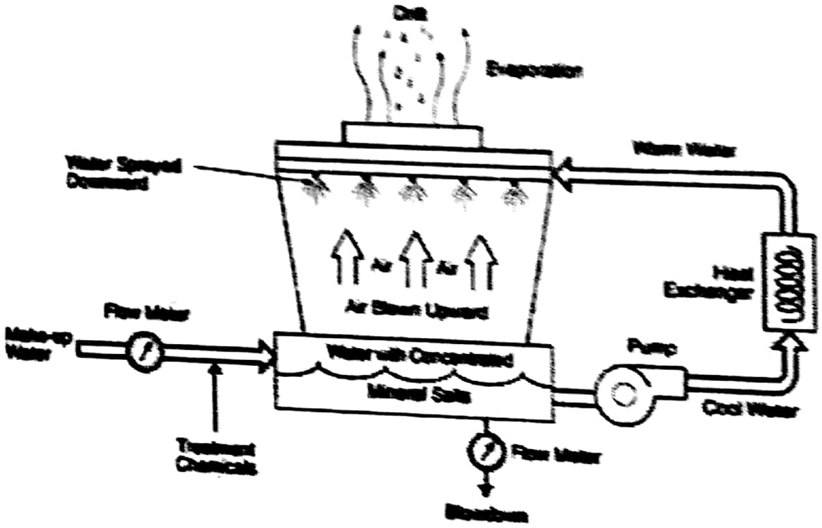
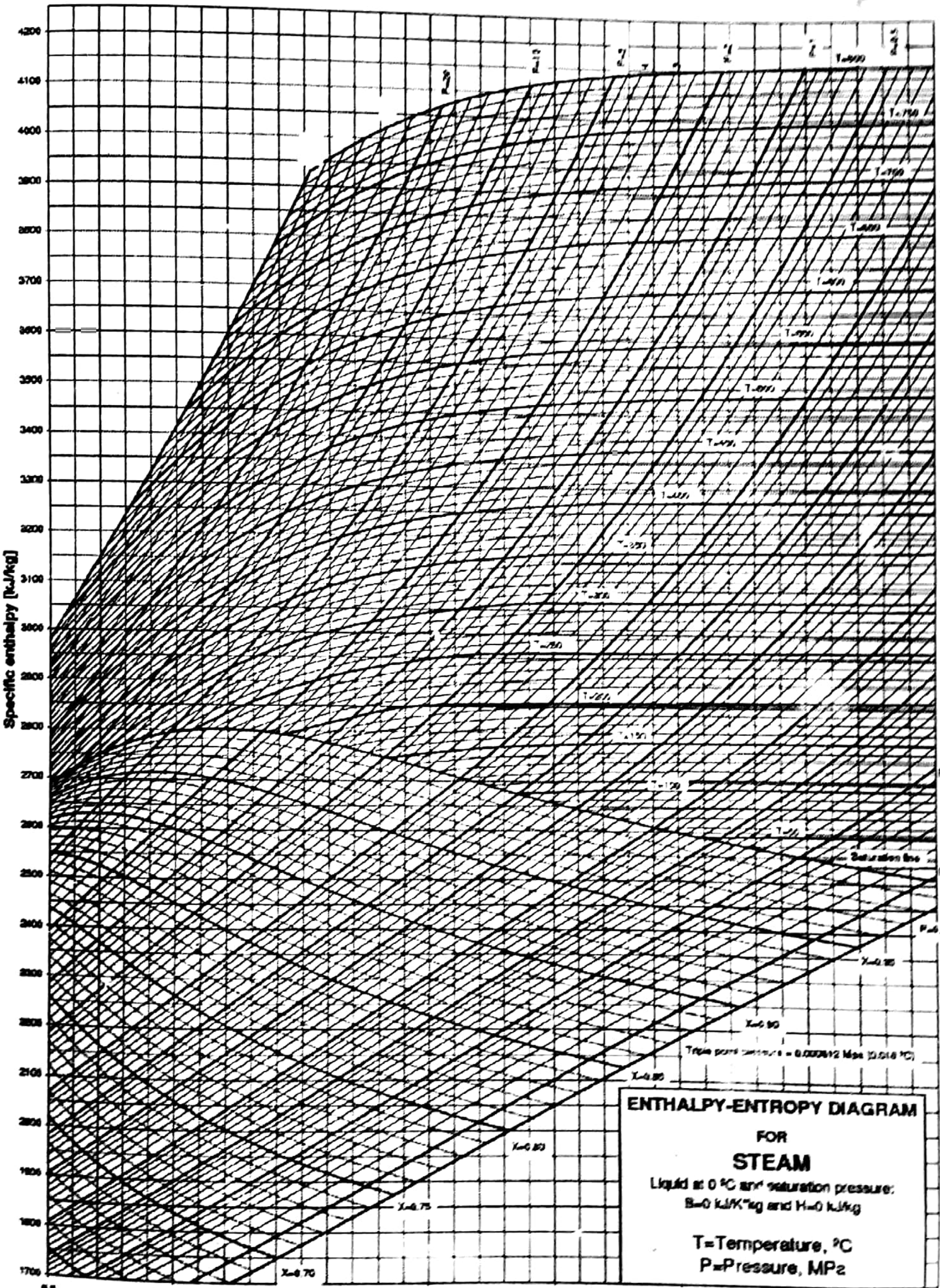


Figure 6.13: Cooling Water Circuit



Produced by I. Aartun, NTNU 2001. Based on the program Allprops, Center for Applied Thermodynamic Studies, University of Idaho.