CHAPTER 2

Vapour Power Cycles

2.1 INTRODUCTION

Steam power plants form the largest contribution for electricity generation. Water is heated in a boiler with the help of a fuel or steam generator in a nuclear reactor to form superheated steam. This high pressure and high temperature steam rotates a steam turbine to produce mechanical work and then electrical power. The steam power plants can operate as:

1. Thermal power plants where chemical energy of a fossil fuel (coal, oil, gas) is released into heat by combustion to raise steam.
2. Nuclear power plants where nuclear reactions of fission or fusion release heat to generate steam.
3. Geothermal power plants where springs of hot water or steam from the interior earth can be directly used to turn a turbine.
4. Solar power plants where solar rays are focused and collected to raise steam.

In ocean thermal power plants, the hot water at the water surface of the sea can be used to run a power plant.

Coal is most abundantly available in India and other countries and is the main fuel for steam power plants.

2.2 THE IDEAL RANKINE CYCLE

In order to overcome most of the limitations of Carnot cycle, the Rankine cycle is used as the basic cycle for steam power plants. An idealized cycle for a simple power plant is shown in Fig. 2.1. The Rankine cycle with superheated steam is 1-2-3-4-1 and that with dry and saturated steam is 1-2′-3′-4-1.

It is made up of four practical processes:

1. Isobaric heating process 1-2: Heat is added to feedwater in three steps:

(i) The water is heated upto its saturated value (process 1-A) in the economizer at constant pressure.

Process 1-2 is reversible and isentropic

$\therefore \quad s_1 = s_2$

At turbine exit, from saturated steam tables,

$$p_2 = 0.05 \text{ bar}$$

$$s_{f2} = 0.476 \text{ kJ/kg}$$

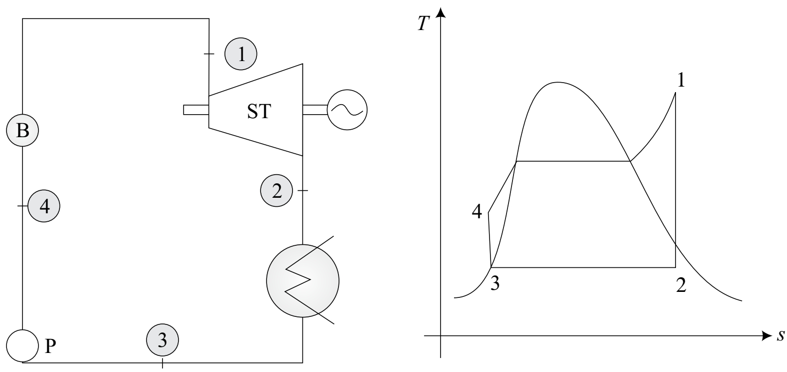
$$s_{fg2} = 7.920 \text{ kJ/kg}$$

$$h_{f2} = 137.80 \text{ kJ/kg}$$

$$h_{fg2} = 2423.7 \text{ kJ/kg}$$

$$s_1 = s_{f2} + x_2 s_{fg2}$$

$$\therefore \quad X_2 = \frac{s_1 - s_{f2}}{s_{fg2}} = \frac{6.756 - 0.476}{7.920} = 0.80.$$



$$\therefore \quad h_2 = h_{f2} + X_2\, h_{fg2} = 137.80 + 0.8\,(2423.7)$$

$$= 2076.76 \text{ kJ/kg.}$$

$\therefore$ Turbine work,

$$W_T = h_1 - h_2 = 3500.9 - 2076.76 = 1424.14 \text{ kJ/kg}$$

(ii) Condenser

At inlet, $h_2 = 2076.76$ kJ/kg

At outlet $h_3 = h_{f_3} = 137.8$ kJ/bg

Heat removed $= h_2 - h_3 = 2076.76 - 137.8 = 1938.96$ kJ/kg

(iii) Pump

At pump inlet, $h_3 = 137.80$ kJ/kg

At pump outlet, $h_4 = h_3 + W_p$

Assuming water as incompressible,

Pump work, $W_p = v_{f3}\,(p_4 - p_3) \times 10^2$

$$W_p = 0.001005\ (100 - 0.05) \times 10^2$$
$$= 10 \text{ kJ/kg}$$

$\therefore$ $$h_4 = 137.80 + 10 = 147.8 \text{ kJ/kg}$$

(iv) Boiler

At inlet of boiler, $h_4 = 147.8$ kJ/kg

At outlet of boiler, $h_1 = 3500.9$ kJ/kg

Heat supplied, $Q_{1-4} = h_1 - h_4 = 3500.9 - 147.8$
$= 3357.1$ kJ/kg

(v) Steam power plant

Cycle efficiency,

$$\eta = \frac{\text{Net work done}}{\text{Heat supplied}}$$

$$= \frac{W_T - W_P}{Q_{1-4}} = \frac{1424.14 - 10}{3353.1} = 0.43$$

$= \mathbf{43\%}$ **Ans**

Specific steam consumption (s.s.c),

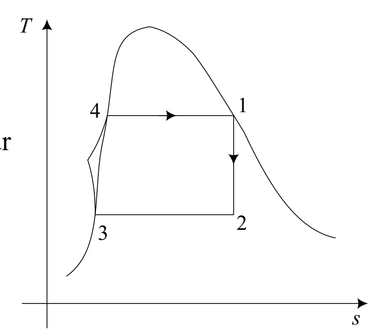
$$\text{s.s.c.} = \frac{3600}{W_T - W_P} = \frac{3600}{1414.14} = 2.55 \text{ kg/kWh}$$

Work Ratio (WR),

$$WR = \frac{W_T - W_P}{W_T} = \frac{1414.14}{1424.14} = \mathbf{0.99} \quad \textbf{Ans}$$

Example 2.2: Steam is supplied to a steam turbine at 5 MPa dry saturated. Condenser pressure is 5 kPa. Showing the Rankine cycle on T-s diagram, determine the simple Rankine efficiency. [U.P.T.U. I Sem., 2004-2005]

Solution:



$$p_1 = 5 \text{ MPa} = 50 \text{ bar}$$
$$p_2 = 5 \text{ kPa} = 0.05 \text{ bar}$$

From saturated steam tables, for $p_1 = 50$ bar

$$h_1 = h_{g_1} = 2794.3 \text{ kJ/kg}$$
$$s_1 = s_{g_1} = 5.973 \text{ kJ/kg-K}$$
$$h_4 = h_{f_4} = 1154.2 \text{ kJ/kg-K}$$

For isentropic expansion through turbine

$$s_1 = s_2 = s_{f2} + X_2\, s_{fg2}$$

From saturated steam tables, for $p_2 = 0.05$ bar

$$s_{f2} = 0.476 \text{ kJ/kg-K}$$

$$s_{fg2} = 7.920 \text{ kJ/kg}$$

$$h_{f2} = 137.8 \text{ kJ/kg}$$

$$h_{fg2} = 2423.7 \text{ kJ/kg}$$

$$\therefore \quad X_2 = \frac{s_1 - s_{f2}}{s_{fg2}} = \frac{5.973 - 0.476}{7.92} = 0.694$$

$$h_2 = h_{f2} + x_2\ h_{fg2} = 137.8 + 0.694\ (2423.7) = 1820 \text{ kJ/kg.}$$

$$\eta_R = \frac{h_1 - h_2}{h_1 - h_4} = \frac{h_1 - h_2}{h_1 - h_3} \qquad \text{(Neglecting pump work)}$$

$$= \frac{2794.3 - 1820}{2794.3 - 137.8} = \frac{974.3}{2656.5} = 36.67\%$$

Example 2.3: A steam power plant working on Rankine cycle has a steam supply pressure of 20 bar and condenser pressure of 0.5 bar. If the initial condition of steam is dry and saturated, calculate the Carnot and Rankine efficiencies of the cycle, neglecting pump work. [U.P.T.U. II Sem., 2005-2006]

Solution:

$$p_1 = 20 \text{ bar}$$

$$p_2 = 0.5 \text{ bar}$$

From steam tables for saturated steam, against a pressure of 20 bar,

$$h_1 = hg_1 = 2799.5 \text{ kJ/kg}$$

$$s_1 = sg_1 = 6.341 \text{ kJ/kg-K.}$$

For isentropic expansion through turbine,

$$s_1 = s_2$$

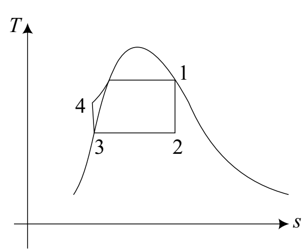
From saturated steam tables at a pressure of 0.5 bar,

$$s_{f2} = 1.091 \text{ kJ/kg-K}$$

$$s_{fg2} = 6.504 \text{ kJ/kg-K}$$

$$h_{f2} = 340.6 \text{ kJ/kg}$$

$$h_{fg2} = 2305.4 \text{ kJ/kg.}$$



$$\therefore \quad s_1 = s_2 = s_{f2} + X_2\, s_{fg2}$$

$$\therefore \quad X_2 = \frac{s_1 - s_{f2}}{s_{fg2}} = \frac{6.341 - 1.091}{6.504} = 0.80$$

$$\therefore \quad h_2 = h_{f2} + X_2 h_{fg2}$$

$$= 340.6 + 0.80\,(2305.4)$$

$$= 1984.92 \text{ kJ/kg}$$

Carnot efficiency,

$$\eta_{\text{Carnot}} = 1 - \frac{T_2}{T_1}$$

From saturated steam tables:

For a pressure of 20 bar,

Saturation temperature, t_{s1}, = 212.4°C

Saturation temperature for 0.5 bar,

$$t_{s2} = 81.35°\text{C}$$

$$T_1 = 212.4 + 273 = 485.4 \text{ K}$$

$$T_2 = 81.35 + 273 = 354.35 \text{ K}$$

$$\therefore \quad \eta_{\text{Carnot}} = 1 - \frac{354.35}{485.4} = 0.27$$

$$= 27\%$$

Neglecting pump work,

$$\eta_{\text{Rankine}} = \frac{h_1 - h_2}{h_1 - h_3} = \frac{2799.5 - 1984.92}{2799.5 - 340.6}$$

$$= \frac{814.58}{2458.9} = 0.33$$

$$= \mathbf{33\%}$$

2.3 THE NON-IDEAL RANKINE CYCLE

There are certain deviations from the assumptions made in the analysis of ideal Rankine cycle.

2.3.1 Thermal Irreversibility in Steam Generation

Heat is added to feed water in three stages.

1. *Economizer:* The water is heated upto saturation temperature (process 4-5) in the economizer at constant pressure. For unit mass flow rate,

$$Q_{ECO} = h_5 - h_4$$

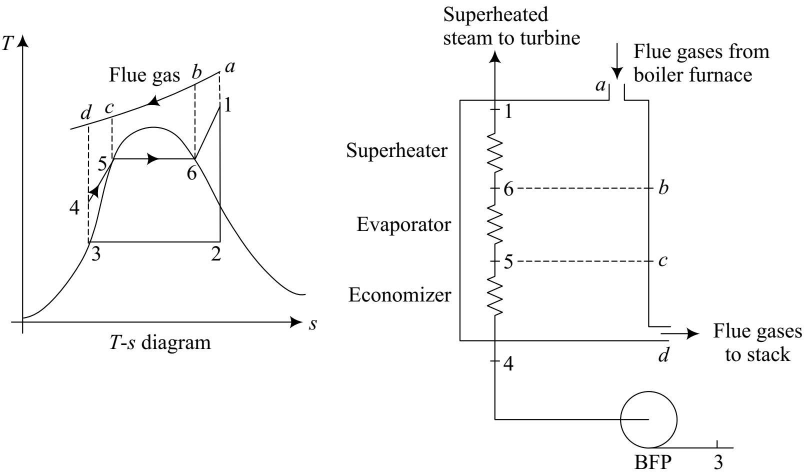


Fig. 2.2 Heat balance of a steam generator

2. *Evaporator:* The saturated water vapour is evaporated to dry saturated steam (process 5-6) in the boiler at constant pressure and temperature.

$$Q_{EVA} = h_6 - h_5 = h_{fg}$$

3. *Superheater:* The saturated steam is superheated at constant pressure (process 6-1) in the superheater.

$$Q_{SH} = h_1 - h_6$$

(a) *Mean temperature of heat supply:* The total heat supplied in the steam generator,

$$\begin{aligned} Q_{\text{in}} &= Q_{ECO} + Q_{EVA} + Q_{SH} \\ &= (h_5 - h_4) + (h_6 - h_5) + (h_1 - h_6) \\ &= (h_1 - h_4) \\ &= T_m (s_1 - s_4) \end{aligned}$$

where T_m = mean temperature of heat addition

$$T_m = \frac{(h_1 - h_4)}{(s_1 - s_4)}$$

The Rankine cycle efficiency,

$$\eta_{\text{Rankine}} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{T_2 (s_1 - s_4)}{T_m (s_1 - s_4)}$$

$$= 1 - \frac{T_2}{T_m}$$

where, T_2 = temperature of heat rejection.

T_2 depends upon condenser pressure but cannot be less than the temperature of surrounding T_∞.

$$\therefore \qquad \eta_{\text{Rankine}} = f(T_m) \text{ only.}$$

The higher the mean temperature of heat addition, the higher will be the cycle efficiency.

(b) *Pinch points:* The working fluid is heated from point 4 to 1 by flue gases getting cooled from a to d in counterflow heat exchangers. The minimum temperature differences are c-5 and 1-a between the two fluids. These points are called *pinch points*. A small pinch point causes increase in surface area and expensive steam generator whereas large pinch point results in lower plant efficiency due to large thermal irreversibility. The most economical pinch-point temperature difference is obtained by optimization to ensure minimum cost of steam generator and minimum operating costs.

2.3.2 Thermal Irreversibility in Condenser

Line 2-3 represents condensation process of steam at constant pressure and temperature.

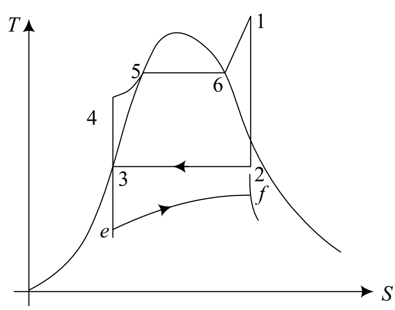


Fig. 2.3 Condenser irreversibility

Line e-f represents the cooling water temperature rise in the condenser. The pinch point f-2 controls the condenser pressure which should be as low as possible and also the size and cost of condenser and pumping cost will increase with the decrease of f-2.

2.3.3 Friction Losses in Steam Turbine and Feed Pump

For ideal Rankine cycle, 1-2′ is the reversible adiabatic and isentropic expansion process in steam turbine and 3-4′ is the isentropic compression in feed pump. The flow rates in turbine and pump are large and the process may be considered adiabatic as heat losses can be neglected. But due to fluid friction, the entropy of fluid will increase in both cases. This is called internal irreversibility and can be expressed by isentropic efficiency.

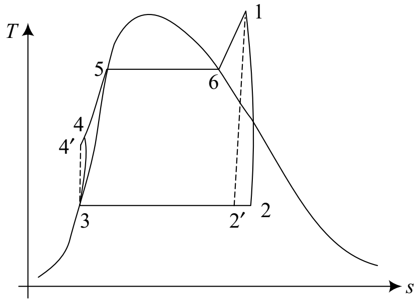


Fig. 2.4 Turbine and pump friction

The isentropic efficiency of steam turbine,

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_2'}$$

The isentropic efficiency of boiler feed pump,

$$\eta_p = \frac{h_4' - h_3}{h_4 - h_3}$$

The actual pump work,

$$W_p = \frac{h_4' - h_3}{\eta_p} = \frac{v_3(p_4 - p_3) \times 10^2}{\eta_p}$$

The turbine produces less work and pump consumes more work due to irreversibility.

2.3.4 Pressure Losses

In an ideal Rankine cycle, p_1 is the pressure at turbine inlet, p_5, the pressure at boiler outlet and p_4, the pressure at pump exit.

$\therefore$ $$p_1 = p_5 = p_4$$

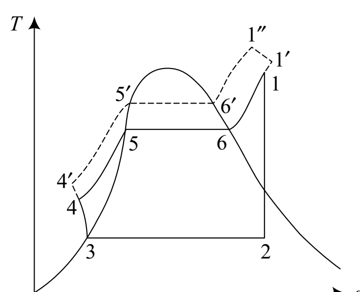


Fig. 2.5 Pressure losses

The liquid leaving the pump must be at a higher pressure (p'_4) than pressure (p_1) at turbine inlet because of pressure drops due to friction in the boiler heat exchangers, feedwater heaters, pipes, bends, valves, etc. In actual cycle $p'_4 > p_5 > p_1$ due to friction in the pipeline and entropy decreases from 1″ to 1 due to heat loss.

2.4 IMPROVEMENT OF CYCLE EFFICIENCY

The Rankine cycle efficiency increases with the increase of mean temperature of heat supply in the cycle. This is achieved by the following methods:

1. Use of high steam parameters, i.e., high pressure and temperature of superheated steam.
2. Reheating of partially expanded steam.
3. Regenerative heating of boiler feed water.

2.4.1 High Steam Parameters

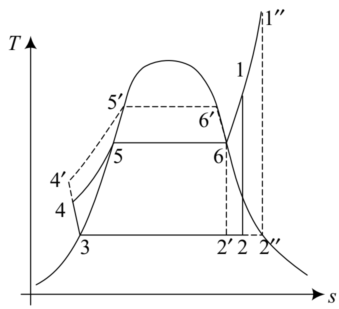


Fig. 2.6 High steam parameters

A higher pressure in the boiler ($p'_5 > p_5$) raises the temperature at which latent heat (h_{fg}) is added which increases mean temperature of heat supply and hence cycle efficiency. However, the moisture content at turbine exhaust ($X_2 > X'_2$) increases which is limited to 12% to avoid blade erosion and other losses.

If the temperature of superheated steam is increased keeping the pressure constant, it has double advantage.

1. Improvement of cycle efficiency due to increase of mean temperature of heat supply.

$$\left[\frac{T_1'' + T_4}{2}\right] > \left[\frac{T_1 + T_4}{2}\right]$$

2. Final wetness in steam is less ($X''_2 > X_2$) which avoids blade erosion.

   The turbine inlet temperature is limited due to metallurgical considerations.

The superheater, piping, valves, steam chest and inlet turbine blades are made from ferrite steels for which temperature limit is 550°C. Austenitic steels have to be used above this temperature which are high alloy steels and four times costlier. For a temperature of 550°C and 12% exhaust wetness, the initial steam pressure would be about 150 bar.

The high steam parameters of the order of 150 bar and 550°C result in the following advantages if used in *large capacity* steam turbines.

1. Extra cost of high temperature components is offset by a general saving in the number of components per MW.
2. Losses are proportionally smaller.
3. High flow rates ensure large and efficient high pressure blading.

2.4.2 Reheating

On large turbines, it becomes economical to increase the cycle efficiency by using reheat, which is a way of partially overcoming the temperature limitations. By returning partially expanded steam from high pressure cylinder to the reheater in the boiler, the mean temperature of heat supply is increased and exhaust wetness is considerably reduced. However, reheating leads to complexity of the plant and higher initial cost. Therefore, reheating is adopted for large turbines above 100 MW.

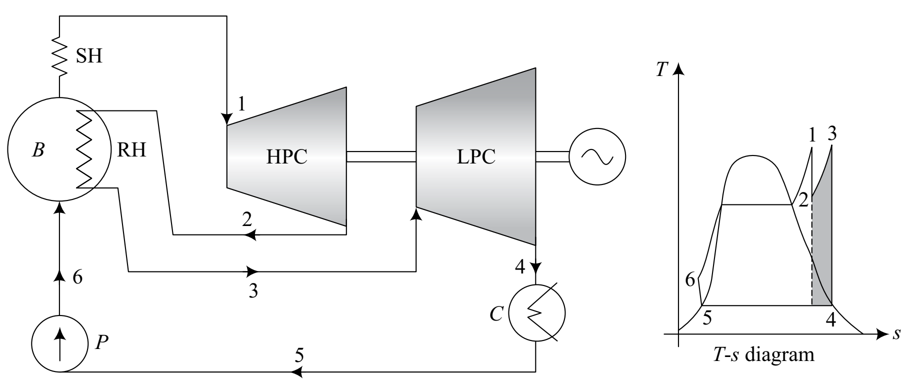


Fig. 2.7 Reheat cycle

2.4.3 Regenerative Heating

Regenerative heating of boiler feedwater is widely used in modern steam power plants which enhances the mean temperature of heat supply in the boiler. The cycle efficiency increases. The amount of heat supplied to the cycle in low temperature region (economizer) by flue gases is reduced. This heating is achieved by steam extracted from turbine.

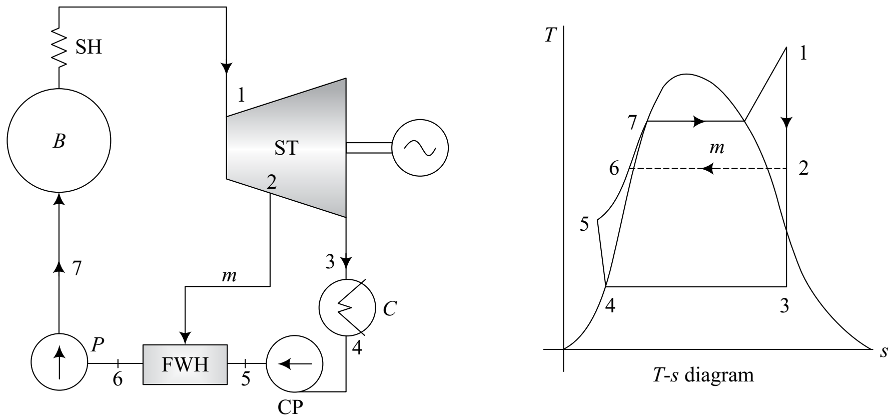


Fig. 2.8 Regenerative heating

2.4.4 Exhaust Pressure

A lower exhaust pressure lowers the temperature at which heat is rejected, which increases cycle efficiency. For condensing turbines, the vacuum obtainable is a function of cooling water temperature at plant site. Any improvement in vacuum is very effective in increasing the work done since a narrow but large addition is made to *T*-*s* diagram area.

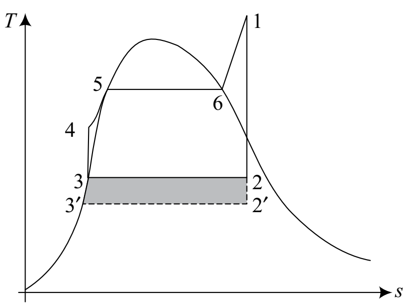


Fig. 2.9 Effect of reduced exhaust pressure

The additional work done = Area 2′-3′-3-2-2′.

2.5 REHEAT CYCLE

A reheat cycle along with *T*-*s* diagram is shown in Fig. 2.10.

The pressure p_2 at which steam is reheated affects the cycle efficiency. The optimum reheat pressure for most of the modern power plants is 0.2 to 0.25 of initial steam pressure.

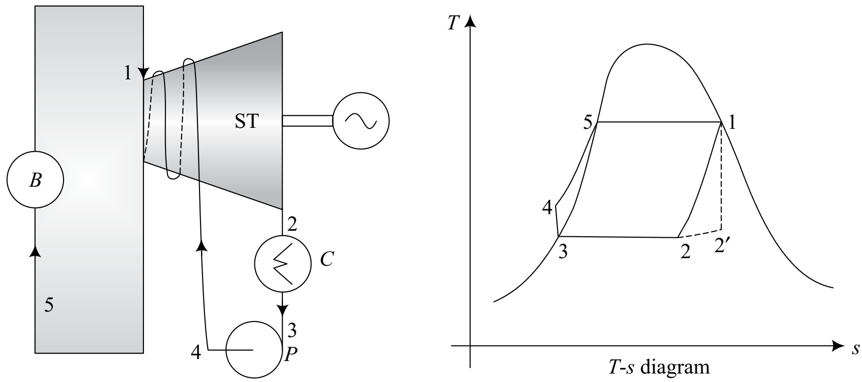


Fig. 2.11 Ideal regeneration cycle

Also, $$s_5 - s_3 = s_1 - s_2$$

∴ $$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2}{T_1}$$

The efficiency of ideal regeneration cycle is equal to that of Carnot cycle.

$$W_T = (h_1 - h_2) - (h_5 - h_4)$$

$$W_p = (h_4 - h_3)$$

$$W_N = W_T - W_p$$

The network output of ideal regeneration cycle is less than that for Rankine cycle.

The ideal regeneration cycle is not practical because:

1. reversible heat transfer cannot be realized;
2. heat exchanger in the turbine is mechanically impractical;
3. the moisture content (X_2) in the turbine is excessive which leads to high erosion of turbine blades.

2.7 REGENERATIVE FEEDWATER HEATING CYCLES

The efficiency of a steam power cycle can be improved by raising the mean temperature at which heat is supplied in the boiler. This can be done in two ways:

1. Increasing the temperature of steam at exit from the boiler. This is done by increasing the pressure of steam, use of superheating and reheating.
2. Increasing the temperature of feedwater at inlet to the boiler. This can be done by preheating water in the economizer by use of exhaust flue gases and regenerative feedwater heating by use of steam bled from the steam turbine at proper points. The use of regenerative feedwater heating results in higher thermal efficiency of the order of 4% but there is decrease in power developed and increase of specific steam consumption (kg/kWh).

There are three types of schemes followed for feedwater heating:

1. Surface heaters with drains cascaded to condenser.
2. Surface heaters with drip pumps.
3. Direct contact heaters.

2.7.1 Surface Heaters with Drains Cascaded to Condenser

This scheme is the simplest and most commonly used in power plants. The feedwater heaters are of shell and tube type heat exchangers. The feedwater passes through the tubes and bled steam, on the shell side. The bled steam is completely condensed. The condensate is fed back to the next lower pressure feedwater heater. The condensate of the lowest pressure feedwater heater is led back to main condenser. The amount of bled steam required for each feedwater heater can be calculated from energy balance that heat gained by feedwater is equal to heat lost by bled steam in condensing.

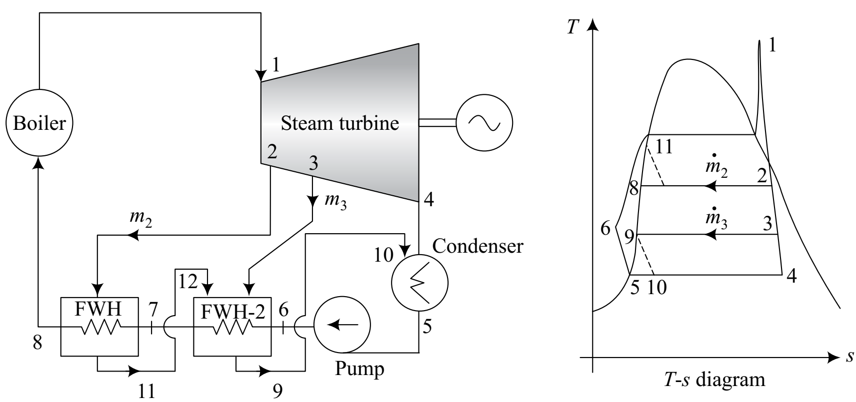


Fig. 2.12 Surface heaters with drains cascaded back to condenser

The system requires throttling of drains before feeding to lower pressure heater and this results in energy loss.

A mass balance based on a unit flow rate at turbine inlet (point 1) is given below:

Mass flow between 1 and 2 = 1

Mass flow between 2 and 3 = $1 - \dot{m}_2$

Mass flow between 3 and 10 = $1 - \dot{m}_2 - \dot{m}_3$

Mass flow between 10 and 1 = 1

Mass flow between 2 and 12 = $\dot{m}_2$

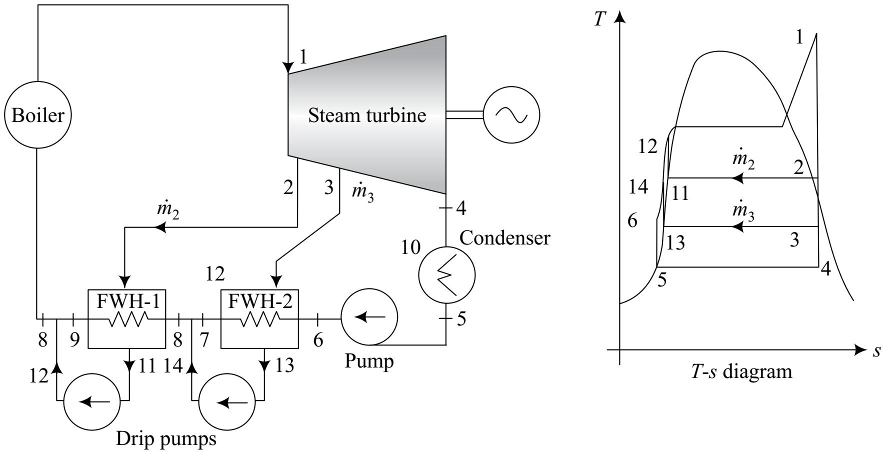


Fig. 2.13 Surface feedwater heaters with drip pumps

A mass balance based on unit mass flow rate at turbine inlet (point 1) is given below:

Mass flow between 1 and 2 $= 1$

Mass flow between 2 and 12 $= \dot{m}_2$

Mass flow between 2 and 3 $= 1 - \dot{m}_2$

Mass flow between 3 and 14 $= \dot{m}_3$

Mass flow between 3 and 7 $= 1 - \dot{m}_2 - \dot{m}_3$

Mass flow at 14 $= \dot{m}_3$

Mass flow between 8 and 9 $= 1 - \dot{m}_2$

Mass flow at 12 $= \dot{m}_2$

Mass flow between 10 and 1 $= 1$

The energy balance on high pressure heater, FWH – 1,

$$\dot{m}_2 (h_2 - h_{11}) = (1 - \dot{m}_2)(h_9 - h_8)$$

The energy balance on low pressure feedwater heater, FWH – 2,

$$\dot{m}_3 (h_3 - h_{13}) = (1 - \dot{m}_2 - \dot{m}_3)(h_7 - h_6)$$

The values of various enthalpies are found out at various temperatures.

$$t_9 = t_{11} - TTD$$

$$t_7 = t_{13} - TTD$$

$$h_{12} = h_{11} + v_{11}\frac{(p_{12} - p_{11})}{\eta_p}$$

$$h_{14} = h_{13} + v_{13}\frac{(p_{14} - p_{13})}{\eta_p}$$

$$h_{10} = \dot{m}_2 h_{12} + (1 - \dot{m}_2)h_9$$

$$(1 - \dot{m}_2)h_8 = \dot{m}_3 h_{14} + (1 - \dot{m}_2 - \dot{m}_3)h_7$$

The turbine work,

$$W_T = (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3) + (1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4)$$

The total pump work,

$$\Sigma W_p = (1 - \dot{m}_2 - \dot{m}_3)(h_6 - h_5) + \dot{m}_3(h_{14} - h_{13}) + \dot{m}_2(h_{12} - h_{11})$$

Heat supplied in the boiler,

$$q_{10-1} = h_1 - h_{10}$$

Thermal efficiency,

$$\eta_{Th} = \frac{W_T - \Sigma W_p}{q_{10-1}}$$

2.7.3 Direct Contact Heaters

The bled steam is mixed directly with the incoming subcooled feedwater to produce saturated water at the extraction pressure. The schematic diagram along with its *T-s* diagram are shown in Fig. 2.14. In addition to condensate pump, as many additional pumps are required as feedwater heaters. Each pump carries full flow of feedwater. Such large pumps increase plant complexity and cost and result in operational, service and noise problems.

In modern power plants only surface type heaters are used. Only one deaerator is used as direct contact heater. The dissolved air in feedwater is removed as water has minimum solubility of air when heated to saturation temperature.

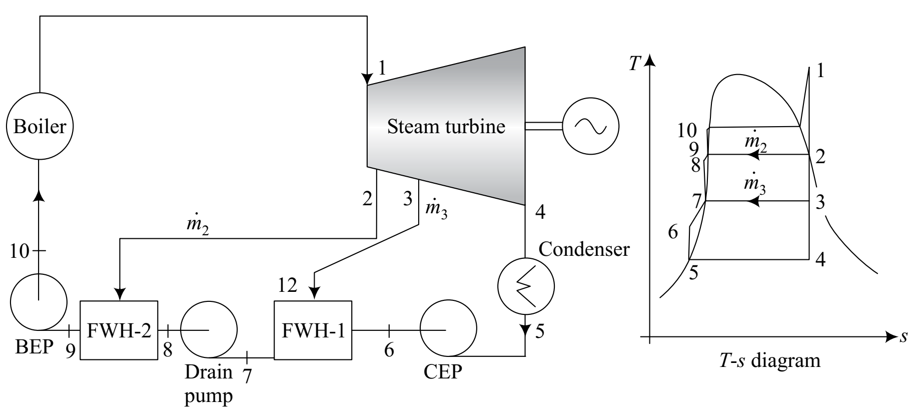


Fig. 2.14 Direct contact heaters