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Power Plant

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Chapter 3

Analyze Steam Power Cycles

Introduction to Steam Tables and Mollier Diagram

1. Definition

Steam is the vapour form of water and is invisible when pure and dry. It does not obey the laws of perfect gases, until it is perfectly dry. When the dry steam is further heated, it behaves more or less like a perfect gas. The steam is, generally, used as a working substance in the operation of steam engines and steam turbines.

2. Formation of steam

The formation of steam takes place, when we continuously heat the water from any temperature in the following stages:

1. The volume of water slightly increases with the increase in temperature. But this increase is, generally, neglected for all types of calculations.
2. On further heating, the temperature of water reaches the boiling point. The boiling temperature of pure steam at normal atmospheric pressure of 1.013 bar (1.013×10^5 N/m^2 or 1.013×10^2 kN/m^2 also known as 1.013×10^5 kPa) is 100°C (373 K). But it increases with the increase in pressure.
3. After boiling point, the temperature remains constant. But the water starts evaporating and the volume of steam starts increasing. At this stage, the steam has some water particles in suspension and it is called wet steam. This process continues, till the whole water is converted into wet steam.
4. On further heating, the temperature of wet steam remains constant and all the water particles in suspension are converted into steam. At this stage, the steam is called dry saturated steam.
5. On further heating, the temperature of dry saturated steam starts increasing and it obeys the laws of perfect gases. At this stage, the steam is called superheated steam. As a matter fact, the superheated steam is used as a working substance in the operation of steam engines and steam turbines.

3. Properties of steam

Thel following properties of steam are always needed for the calculations of its various parameters, which are required in the operation of steam engines and steam turbines.

1. *Specific volume of steam.* It is the volume occupied by the steam per unit mass at a given temperature and pressure. It is expressed in m^3/kg and is the reciprocal of the density of steam. The specific volume of steam increases with the increase in temperature and decreases with the increase in pressure.
2. *Specific enthalpy of steam.* It is the total heat absorbed by the steam per unit mass from the freezing point of water (0°C or 273 K) to the saturation temperature (100°C or 373 K) plus the heat absorbed during evaporation. It is expressed in kJ/kg. The specific enthalpy of steam increases with the increase in temperature and pressure.

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3. *Specific entropy of steam.* It is a theoretical value of heat energy, which can not be transformed into mechanical work under the given conditions of temperature or pressure. It is also called degree of disorder of the system. The most common term used is the change of entropy, which is mathematically given as :

$$\Delta s = \frac{\Delta Q}{\Delta T} = \frac{\text{Heat supplied}}{\text{Temperature of the system}}$$

It is expressed in kJ/kg K. The specific entropy of steam decreases with an increase in temperature and pressure.

4. Steam tables

The various properties of steam (such as specific volume, specific enthalpy and specific entropy) of dry saturated steam and superheated steam vary with the variations of temperature and pressure. These values were carefully determined by observations and calculations first in F.P.S. system and were made available in a tabular form known as steam tables. Later on, these values were converted into M.K.S. units and then into S.I. units. Due to conversion and rounding off the figures, there is a slight difference in the figures quoted in different books. Even some of the authors have changed these values in different editions of the same book. However, in this steam tables the author has quoted standard figures, which are widely accepted and internationally recognised.

There are two important steam tables. One of them is based in terms of temperature and the other in terms of pressure. It is a general practice to give the following tables for some important values:

1. Saturated water and steam (temperature) table
2. Saturated water and steam (pressure) table
3. Superheated steam table
4. Supercritical steam table

5. Saturated water and steam (temperature) table

It contains values of absolute pressure (in bar), specific volume (in m^3/kg), specific enthalpy (in kJ/kg) and specific entropy (in kJ/kg K) from 0°C to 374.15°C (critical temperature). A sample of this table is given below:

Temperature in °C (t)	Absolute Pressure in bar (p)	Specific volume in m^3/kg		Specific enthalpy in kJ/kg			Specific entropy in kJ/kg K		
		Water (v_f)	Steam (v_g)	Water (h_f)	Evaporation (h_{fg})	Steam (h_g)	Water (s_f)	Evaporation (s_{fg})	Steam (s_g)
0	0.006 11	0.001 000	206.16	0.0	2501.6	2501.6	0.000	9.158	9.158
5	0.008 72	0.001 000	147.16	21.0	2489.7	2510.7	0.076	8.951	9.027
10	0.012 27	0.001 000	106.43	42.0	2477.9	2519.9	0.151	8.751	8.902

The use of this table is given in the following example.

Example 1. Calculate the specific enthalpy and specific entropy of 1 kg of steam at 10°C when its dryness fraction is 0.8.

Solution. Given: Mass of steam $(m) = 1$ kg; Temperature of steam $(t) = 10$°C and dryness fraction of steam $(x) = 0.8$.

From steam tables, corresponding to a temperature of 10°C, we find that $h_f = 42.0$ kJ/kg; $h_{fg} = 2477.9$ kJ/kg; $s_f = 0.151$ kJ/kg K and $s_{fg} = 8.751$ kJ/kg K.

Specific enthalpy of the steam

We know that specific enthalpy of the steam,

$$h = m\,[h_f + x\,h_{fg}] = 1 \times [42.0 + (0.8 \times 2477.9)] = 2024.32 \text{ kJ} \quad \textbf{Ans.}$$

Specific entropy of the steam

We also know that specific entropy of the steam,

$$s = m\,[s_f + x\,s_{fg}] = 1 \times [0.151 + (0.8 \times 8.751)] = 7.1518 \text{ kJ/kg K} \quad \textbf{Ans.}$$

6. Staturated water and steam (pressure) tables

It contains the values of temperature (in °C), specific volume (in m³/kg), specific enthalpy (in kJ/kg) and specific entropy (in kJ / kg K) from 0.0061 bar to 221.2 bar (critical pressure). A sample of this table is given below:

Absolute Pressure in bar (p)	Tempe-rature in °C (t)	Specific volume in m³/kg		Specific enthalpy in kJ/kg			Specific entropy in kJ/kg K		
		Water (v_f)	Steam (v_g)	Water (h_f)	Evapo-ration (h_{fg})	Steam (h_g)	Water (s_f)	Evapora-tion (s_{fg})	Steam (s_g)
0.010	6.98	0.001 000	129.21	29.3	2485.0	2514.4	0.106	8.871	8.977
0.020	17.51	0.001 001	67.012	73.5	2460.2	2533.6	0.261	8.464	8.725
0.030	24.10	0.001 003	45.670	101.0	2444.6	2545.6	0.354	8.224	8.578

The use of this table is given in the following example.

Example 2. What is the specific enthalpy and specific entropy of 1.5 kg of steam at a pressure of 0.030 bar, when its dryness fraction is 0.6?

Solution. Given: Mass of steam $(m) = 1.5$ kg; Pressure of steam $= 0.030$ bar and dryness fraction of steam $(x) = 0.6$.

From steam tables, corresponding to a pressure of 0.030 bar, we find that $h_f = 101$ kJ/kg; $h_{fg} = 2444.6$ kJ/kg; $s_f = 0.354$ kJ/kg K and $s_{fg} = 8.224$ kJ/kg K.

Specific enthalpy of the steam

We know that specific enthalpy of the steam,

$$h = m\,[h_f + x\,h_{fg}] = 1 \times [101 + (0.6 \times 2444.6)] = 1567.76 \text{ kJ} \quad \textbf{Ans.}$$

1.5

Specific entropy of the steam

We also know that specific entropy of the steam,

$$s = m\,[s_f + x\,s_{fg}] = 1.5 \times [0.354 + (0.6 \times 8.224)] = 7.9326 \text{ kJ/kg K} \quad \textbf{Ans.}$$

7. Superheated steam tables

These tables contain the values of specific volume, specific enthalpy and specific entropy of superheated steam from an absolute pressure of 0.02 bar to 221.2 bar (critical pressure) at various temperatures from 100°C to 800°C. In these tables, the values of specific volume, specific enthalpy and specific entropy of steam are directly read from the concerned tables. However, the value at any other pressure or temperature, not mentioned in the tables, is obtained by interpolation.

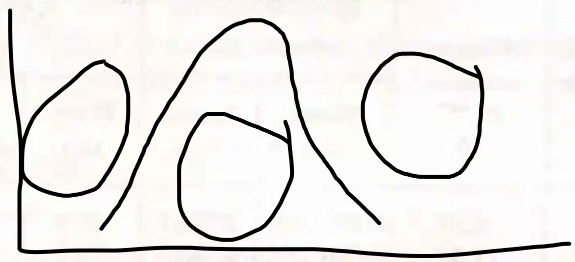
8. Supercritical steam tables

These tables also contain the values of specific volume, specific enthalpy and specific entropy of supercritical steam from an absolute pressure of 250 bar to 1000 bar at various temperatures from 400°C to 800°C. In these tables also, the values of specific volume, specific enthalpy and specific entropy of steam are directly read from the concerned tables. However, the value at any other pressure or temperature, not mentioned in the table, is obtained by interpolation.

9. Mollier diagram

It is a graphical representation of steam tables, in which specific entropy is plotted along the ordinate (X-axis) and specific enthalpy along the abscissa (Y-axis). The diagram is divided into two portions by a somewhat horizontal line termed as saturation curve. The lower portion (*i.e.*, wet steam region) contains the values of wet steam, whereas the upper portion (*i.e.*, superheated steam region) contains the values of superheated steam. A Mollier diagram has the following lines.

1. Dryness fraction lines
2. Constant specific volume lines
3. Constant pressure lines
4. Constant temperature lines



10. Dryness fraction lines

These lines are drawn in the wet steam region. *i.e.*, only below the saturation curve (which represents dryness fraction equal to unity). These lines represent the condition of wet steam between various values of enthalpy and entropy. The dryness fraction lines are slightly curved in horizontal direction.

11. Constant specific volume lines

These lines are drawn in both the wet steam region and superheated steam region. These lines represent the specific volume of steam between the various values of enthalpy and entropy. The lines are straight in the wet steam region, *i.e.*, below the saturation curve, but are curved upwards in the superheated region *i.e.*, above the saturation curve.

12. Constant pressure lines

These lines are also drawn in both the wet steam region and superheated steam region. These lines represent the pressure of steam between the various values of enthalpy and entropy. The pressure lines are also straight in the wet steam region, *i.e.*, below the saturation curve, but are curved slightly upwards in the superheated region *i.e.*, above the saturation curve.

13. Constant temperature lines

These lines are drawn only in the superheated steam region *i.e.*, above the saturation curve. These lines represent the temperature of steam between the various values of enthalpy and entropy The temperature lines are slightly curved in the horizontal direction.

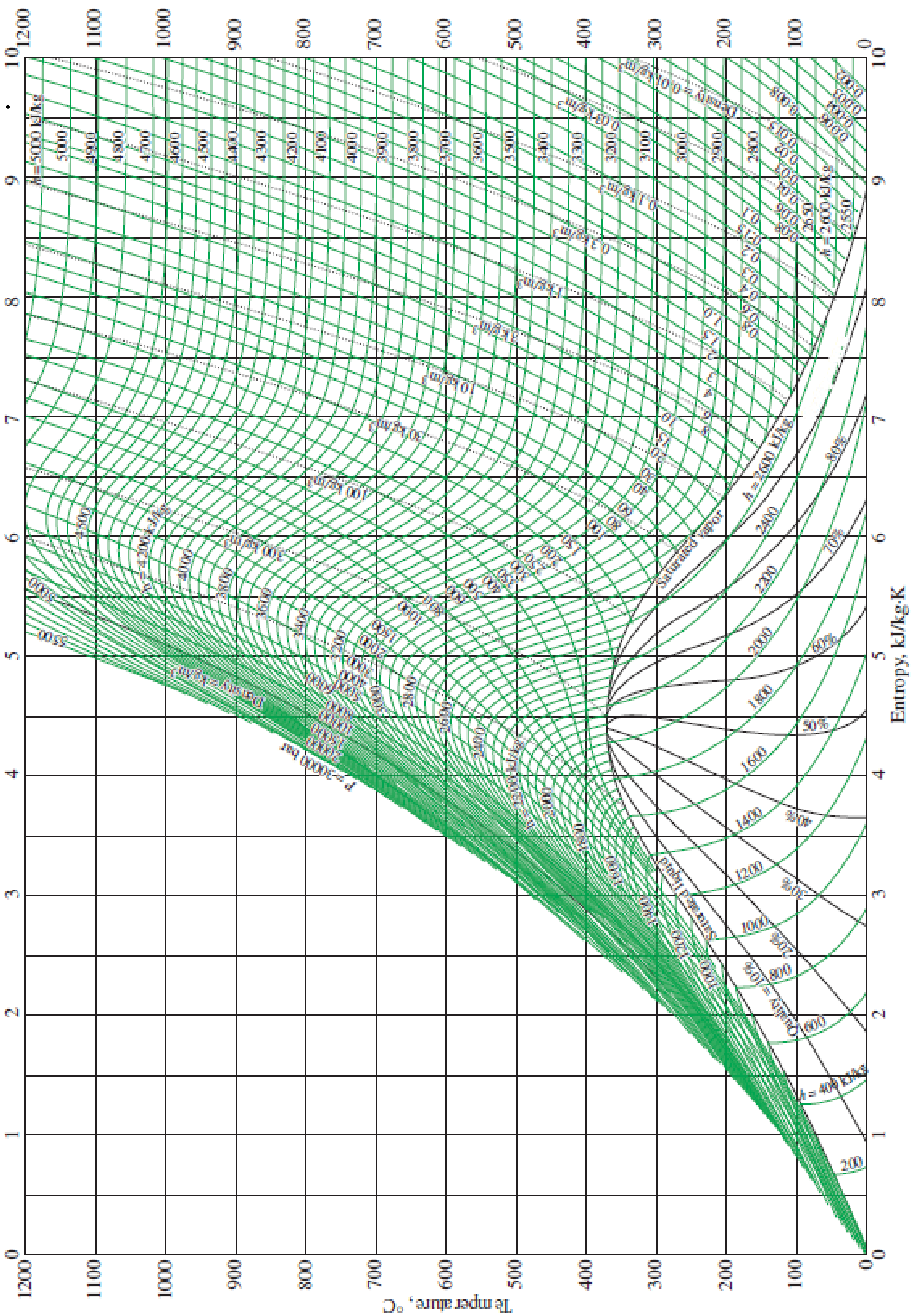


FIGURE A–9

T-s diagram for water.

Source of Data: From NBS/NRC Steam Tables/1 *by Lester Haar, John S. Gallagher, and George S. Kell. Routledge/Taylor & Francis Books, Inc.,* 1984.

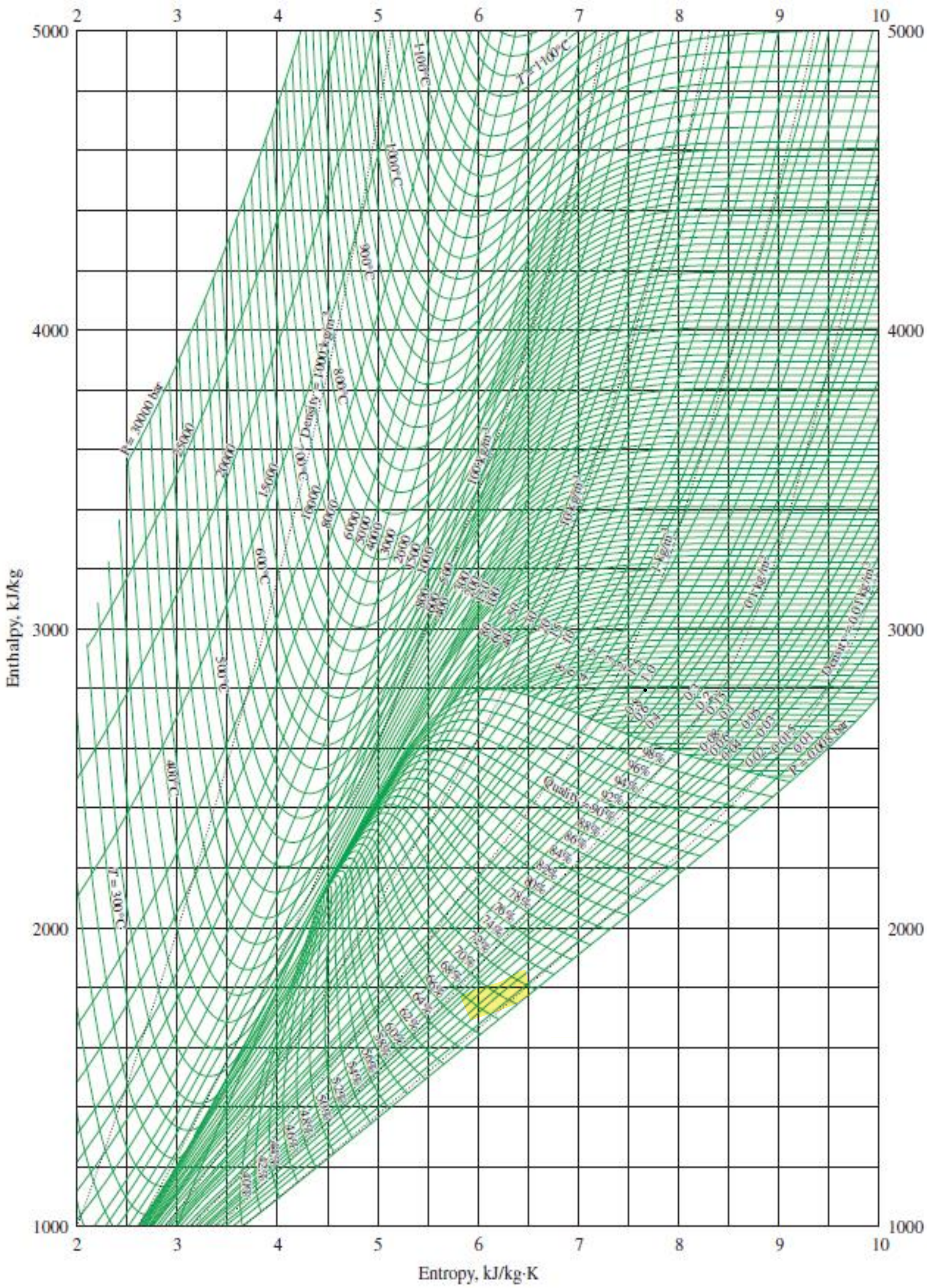


FIGURE A-10

Mollier diagram for water.

Source of Data: From NBS/NRC Steam Tables/1 *by Lester Haar, John S. Gallagher, and George S. Kell. Routledge/Taylor & Francis Books, Inc.*, 1984.

RANKINE CYCLE: THE IDEAL CYCLE FOR VAPOR POWER CYCLES

Many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser, as shown schematically on a *T-s* diagram in Fig. 10–2. The cycle that results is the **Rankine cycle,** which is the ideal cycle for vapor power plants. The ideal Rankine cycle does not involve any internal irreversibilities and consists of the following four processes:

1-2 Isentropic compression in a pump
2-3 Constant pressure heat addition in a boiler
3-4 Isentropic expansion in a turbine
4-1 Constant pressure heat rejection in a condenser

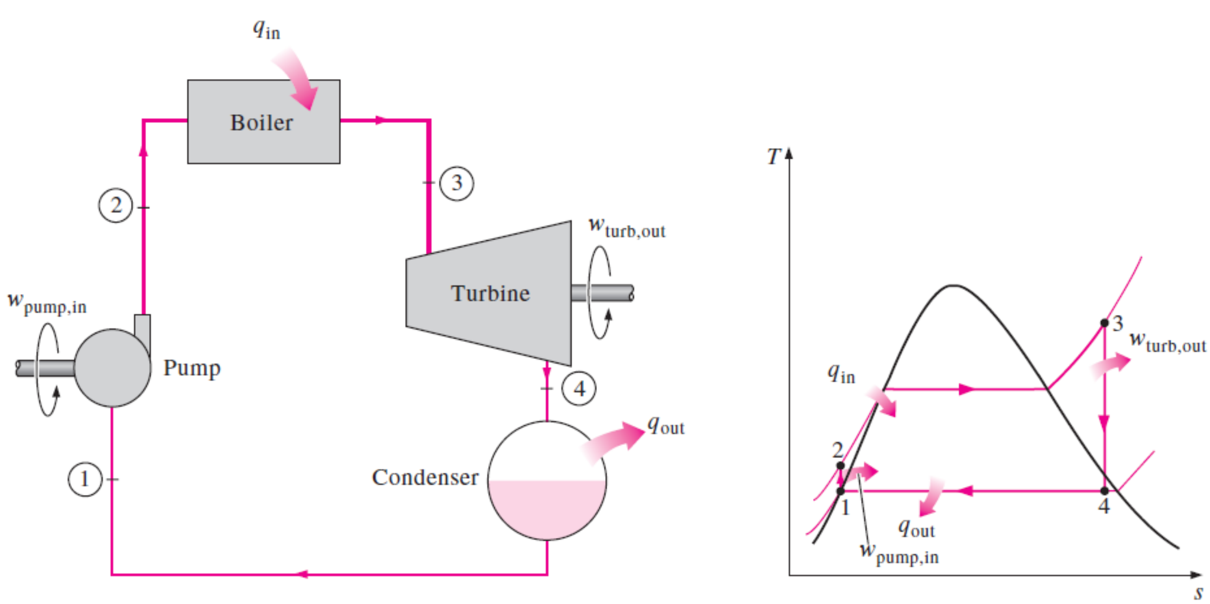


FIGURE 10–2
The simple ideal Rankine cycle.

Energy Analysis of the Ideal Rankine Cycle

All four components associated with the Rankine cycle (the pump, boiler, turbine, and condenser) are steady-flow devices, and thus all four processes that make up the Rankine cycle can be analyzed as steady-flow processes. The kinetic and potential energy changes of the steam are usually small relative to the work and heat transfer terms and are therefore usually neglected. Then the *steady-flow energy equation* per unit mass of steam reduces to

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i \qquad \text{(kJ/kg)} \qquad (10\text{–}1)$$

The boiler and the condenser do not involve any work, and the pump and the turbine are assumed to be isentropic. Then the conservation of energy relation for each device can be expressed as follows:

Pump ($q = 0$): $$w_{pump,in} = h_2 - h_1 \qquad (10\text{–}2)$$

or,

$$w_{pump,in} = v(P_2 - P_1) \qquad (10\text{–}3)$$

where

$$h_1 = h_{f\,@\,P_1} \quad \text{and} \quad v \cong v_1 = v_{f\,@\,P_1} \qquad (10\text{–}4)$$

Boiler ($w = 0$): $$q_{in} = h_3 - h_2 \qquad (10\text{–}5)$$

Turbine ($q = 0$): $$w_{turb,out} = h_3 - h_4 \qquad (10\text{–}6)$$

Condenser ($w = 0$): $$q_{out} = h_4 - h_1 \qquad (10\text{–}7)$$

The *thermal efficiency* of the Rankine cycle is determined from

$$\eta_{th} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} \qquad (10\text{–}8)$$

where

$$w_{net} = q_{in} - q_{out} = w_{turb,out} - w_{pump,in}$$

تحويل

The conversion efficiency of power plants in the United States is often expressed in terms of **heat rate,** which is the amount of heat supplied, in Btu's, to generate 1 kWh of electricity. The smaller the heat rate, the greater the efficiency. Considering that 1 kWh = 3412 Btu and disregarding the losses associated with the conversion of shaft power to electric power, the relation between the heat rate and the thermal efficiency can be expressed as

$$\eta_{th} = \frac{3412\ \text{(Btu/kWh)}}{\text{Heat rate (Btu/kWh)}} \qquad (10\text{–}9)$$

For example, a heat rate of 11,363 Btu/kWh is equivalent to 30 percent efficiency.

The thermal efficiency can also be interpreted as the ratio of the area enclosed by the cycle on a *T-s* diagram to the area under the heat-addition process. The use of these relations is illustrated in the following example.

EXAMPLE 10–2 An Actual Steam Power Cycle

A steam power plant operates on the cycle shown in Fig. 10–5. If the isentropic efficiency of the turbine is 87 percent and the isentropic efficiency of the pump is 85 percent, determine (*a*) the thermal efficiency of the cycle and (*b*) the net power output of the plant for a mass flow rate of 15 kg/s.

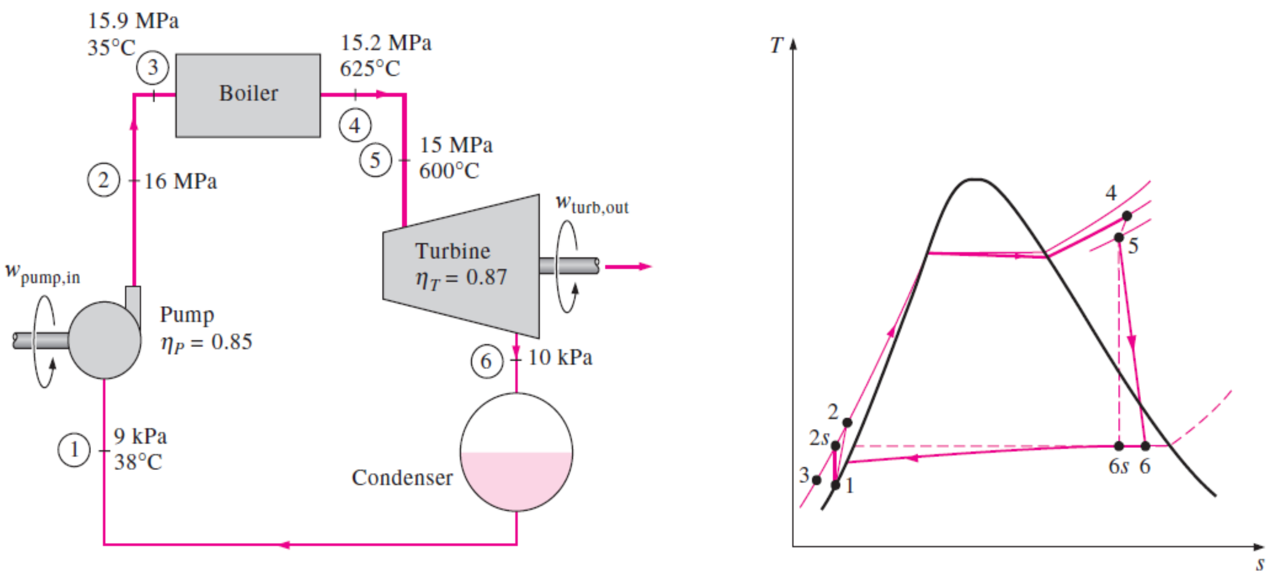


FIGURE 10–5
Schematic and *T-s* diagram for Example 10–2.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–5. The temperatures and pressures of steam at various points are also indicated on the figure. We note that the power plant involves steady-flow components and operates on the Rankine cycle, but the imperfections at various components are accounted for.

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(*a*) The thermal efficiency of a cycle is the ratio of the net work output to the heat input, and it is determined as follows:

Pump work input:

$$w_{pump,in} = \frac{w_{s,pump,in}}{\eta_P} = \frac{v_1(P_2 - P_1)}{\eta_P}$$

$$= \frac{(0.001009\ m^3/kg)[(16{,}000 - 9)\ kPa]}{0.85}\left(\frac{1\ kJ}{1\ kPa \cdot m^3}\right)$$

$$= 19.0\ kJ/kg$$

Turbine work output:

$$w_{turb,out} = \eta_T w_{s,turb,out}$$
$$= \eta_T(h_5 - h_{6s}) = 0.87(3583.1 - 2115.3)\ \text{kJ/kg}$$
$$= 1277.0\ \text{kJ/kg}$$

Boiler heat input: $q_{in} = h_4 - h_3 = (3647.6 - 160.1)\ \text{kJ/kg} = 3487.5\ \text{kJ/kg}$

Thus,

$$w_{net} = w_{turb,out} - w_{pump,in} = (1277.0 - 19.0)\ \text{kJ/kg} = 1258.0\ \text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{1258.0\ \text{kJ/kg}}{3487.5\ \text{kJ/kg}} = \mathbf{0.361\ or\ 36.1\%}$$

(*b*) The power produced by this power plant is

$$\dot{W}_{net} = \dot{m}(w_{net}) = (15\ \text{kg/s})(1258.0\ \text{kJ/kg}) = \mathbf{18.9\ MW}$$

Discussion Without the irreversibilities, the thermal efficiency of this cycle would be 43.0 percent (see Example 10–3*c*).

To here 4-3-2026

10–4 ▪ HOW CAN WE INCREASE THE EFFICIENCY OF THE RANKINE CYCLE?

Steam power plants are responsible for the production of most electric power in the world, and even small increases in thermal efficiency can mean large savings from the fuel requirements. Therefore, every effort is made to improve the efficiency of the cycle on which steam power plants operate.

The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same: *Increase the average temperature at which heat is transferred to the working fluid in the boiler,* or *decrease the average temperature at which heat is rejected from the working fluid in the condenser.* That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection. Next we discuss three ways of accomplishing this for the simple ideal Rankine cycle.

Lowering the Condenser Pressure (*Lowers* $T_{low,avg}$)

Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. Therefore, lowering the operating pressure of the condenser automatically lowers the temperature of the steam, and thus the temperature at which heat is rejected.

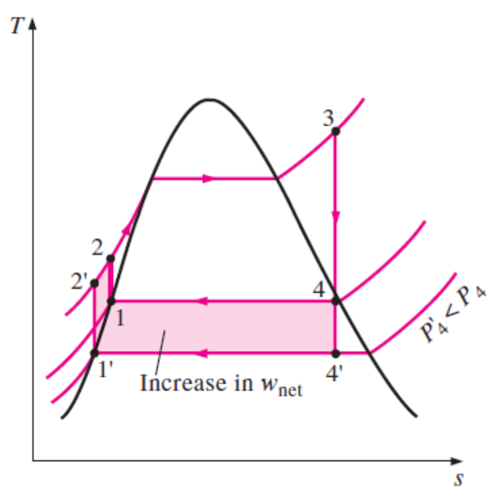


FIGURE 10–6

The effect of lowering the condenser pressure on the ideal Rankine cycle.

Superheating the Steam to High Temperatures (*Increases* $T_{high,avg}$)

The average temperature at which heat is transferred to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapor power cycles is illustrated on a *T-s* diagram in Fig. 10–7. The colored area on

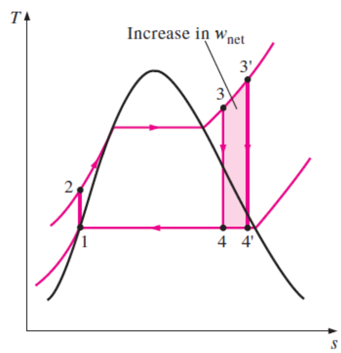


FIGURE 10–7

The effect of superheating the steam to higher temperatures on the ideal Rankine cycle.

Increasing the Boiler Pressure (*Increases* $T_{high,avg}$)

Another way of increasing 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 takes place. This, in turn, raises the average temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle.

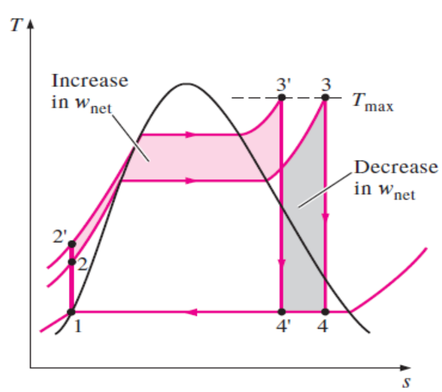


FIGURE 10–8

The effect of increasing the boiler pressure on the ideal Rankine cycle.

EXAMPLE 10–3 Effect of Boiler Pressure and Temperature on Efficiency

Consider a steam power plant operating on the ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350°C and is condensed in the condenser at a pressure of 10 kPa. Determine (*a*) the thermal efficiency of this power plant, (*b*) the thermal efficiency if steam is superheated to 600°C instead of 350°C, and (*c*) the thermal efficiency if the boiler pressure is raised to 15 MPa while the turbine inlet temperature is maintained at 600°C.

Solution A steam power plant operating on the ideal Rankine cycle is considered. The effects of superheating the steam to a higher temperature and raising the boiler pressure on thermal efficiency are to be investigated.

Analysis The *T-s* diagrams of the cycle for all three cases are given in Fig. 10–10.

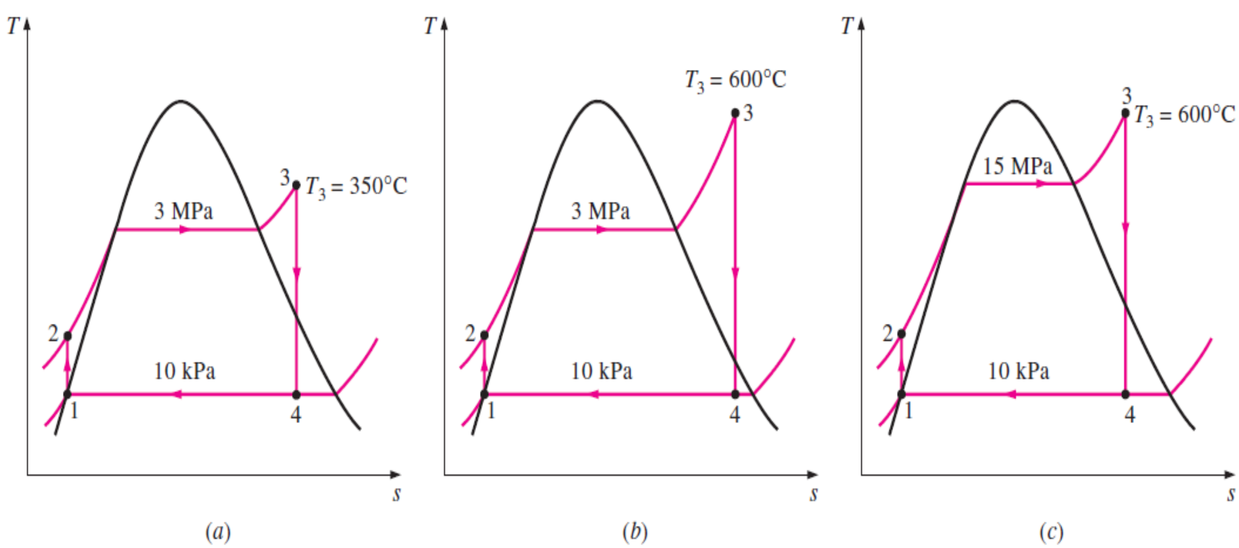


FIGURE 10–10

T-s diagrams of the three cycles discussed in Example 10–3.

(*a*) This is the steam power plant discussed in Example 10–1, except that the condenser pressure is lowered to 10 kPa. The thermal efficiency is determined in a similar manner:

State 1: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \quad \begin{array}{l} h_1 = h_{f\,@\,10\text{ kPa}} = 191.81 \text{ kJ/kg} \\ v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3/\text{kg} \end{array}$

State 2: $P_2 = 3$ MPa
$s_2 = s_1$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg})[(3000 - 10) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= 3.02 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 3.02) \text{ kJ/kg} = 194.83 \text{ kJ/kg}$$

State 3: $\left.\begin{array}{l} P_3 = 3 \text{ MPa} \\ T_3 = 350°\text{C} \end{array}\right\} \quad \begin{array}{l} h_3 = 3116.1 \text{ kJ/kg} \\ s_3 = 6.7450 \text{ kJ/kg}\cdot\text{K} \end{array}$

State 4: $P_4 = 10$ kPa (sat. mixture)
$s_4 = s_3$

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7450 - 0.6492}{7.4996} = 0.8128$$

Thus,

$$h_4 = h_f + x_4 h_{fg} = 191.81 + 0.8128(2392.1) = 2136.1 \text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = (3116.1 - 194.83) \text{ kJ/kg} = 2921.3 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = (2136.1 - 191.81) \text{ kJ/kg} = 1944.3 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1944.3 \text{ kJ/kg}}{2921.3 \text{ kJ/kg}} = \mathbf{0.334 \text{ or } 33.4\%}$$

Therefore, the thermal efficiency increases from 26.0 to 33.4 percent as a result of lowering the condenser pressure from 75 to 10 kPa. At the same time, however, the quality of the steam decreases from 88.6 to 81.3 percent (in other words, the moisture content increases from 11.4 to 18.7 percent).

(*b*) States 1 and 2 remain the same in this case, and the enthalpies at state 3 (3 MPa and 600°C) and state 4 (10 kPa and $s_4 = s_3$) are determined to be

$$h_3 = 3682.8 \text{ kJ/kg}$$

$$h_4 = 2380.3 \text{ kJ/kg} \quad (x_4 = 0.915)$$

Thus,

$$q_{in} = h_3 - h_2 = 3682.8 - 194.83 = 3488.0 \text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 2380.3 - 191.81 = 2188.5 \text{ kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2188.5 \text{ kJ/kg}}{3488.0 \text{ kJ/kg}} = \mathbf{0.373 \text{ or } 37.3\%}$$

Therefore, the thermal efficiency increases from 33.4 to 37.3 percent as a result of superheating the steam from 350 to 600°C. At the same time, the quality of the steam increases from 81.3 to 91.5 percent (in other words, the moisture content decreases from 18.7 to 8.5 percent).

(*c*) State 1 remains the same in this case, but the other states change. The enthalpies at state 2 (15 MPa and $s_2 = s_1$), state 3 (15 MPa and 600°C), and state 4 (10 kPa and $s_4 = s_3$) are determined in a similar manner to be

$$h_2 = 206.95 \text{ kJ/kg}$$

$$h_3 = 3583.1 \text{ kJ/kg}$$

$$h_4 = 2115.3 \text{ kJ/kg} \quad (x_4 = 0.804)$$

Thus,

$$q_{in} = h_3 - h_2 = 3583.1 - 206.95 = 3376.2 \text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 2115.3 - 191.81 = 1923.5 \text{ kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1923.5 \text{ kJ/kg}}{3376.2 \text{ kJ/kg}} = \mathbf{0.430 \text{ or } 43.0\%}$$

To here 8/3/2026

10–5 ▪ THE IDEAL REHEAT RANKINE CYCLE

We noted in the last section that increasing the boiler pressure increases the thermal efficiency of the Rankine cycle, but it also increases the moisture content of the steam to unacceptable levels. Then it is natural to ask the following question:

How can we take advantage of the increased efficiencies at higher boiler pressures without facing the problem of excessive moisture at the final stages of the turbine?

Two possibilities come to mind:

1. Superheat the steam to very high temperatures before it enters the turbine. This would be the desirable solution since the average temperature at which heat is added would also increase, thus increasing the cycle efficiency. This is not a viable solution, however, since it requires raising the steam temperature to metallurgically unsafe levels.
2. Expand the steam in the turbine in two stages, and reheat it in between. In other words, modify the simple ideal Rankine cycle with a **reheat** process. Reheating is a practical solution to the excessive moisture problem in turbines, and it is commonly used in modern steam power plants.

$$q_{in} = q_{primary} + q_{reheat} = (h_3 - h_2) + (h_5 - h_4) \quad (10\text{–}12)$$

and

$$w_{turb,out} = w_{turb,I} + w_{turb,II} = (h_3 - h_4) + (h_5 - h_6) \quad (10\text{–}13)$$

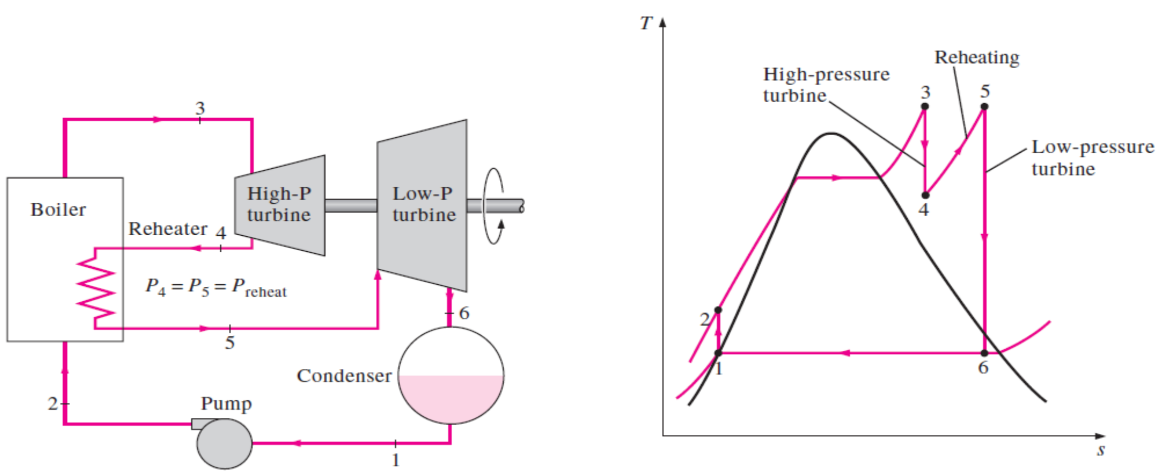


FIGURE 10–11
The ideal reheat Rankine cycle.

EXAMPLE 10–4 The Ideal Reheat Rankine Cycle

Consider a steam power plant operating on the ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. If the moisture content of the steam at the exit of the low-pressure turbine is not to exceed 10.4 percent, determine (*a*) the pressure at which the steam should be reheated and (*b*) the thermal efficiency of the cycle. Assume the steam is reheated to the inlet temperature of the high-pressure turbine.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–13. We note that the power plant operates on the ideal reheat Rankine cycle. Therefore, the pump and the turbines are isentropic, there are no pressure drops in the boiler and condenser, and steam leaves the condenser and enters the pump as saturated liquid at the condenser pressure.

(*a*) The reheat pressure is determined from the requirement that the entropies at states 5 and 6 be the same:

State 6: $P_6 = 10 \text{ kPa}$

$$x_6 = 0.896 \quad (\text{sat. mixture})$$ 1 - 0.104 = 0.896

$$s_6 = s_f + x_6 s_{fg} = 0.6492 + 0.896(7.4996) = 7.3688 \text{ kJ/kg} \cdot \text{K}$$

Also,

$$h_6 = h_f + x_6 h_{fg} = 191.81 + 0.896(2392.1) = 2335.1 \text{ kJ/kg}$$

Thus,

State 5:

$$\left.\begin{array}{l} T_5 = 600°\text{C} \\ s_5 = s_6 \end{array}\right\} \quad \begin{array}{l} P_5 = \mathbf{4.0\ MPa} \\ h_5 = 3674.9 \text{ kJ/kg} \end{array}$$

Therefore, steam should be reheated at a pressure of 4 MPa or lower to prevent a moisture content above 10.4 percent.

(*b*) To determine the thermal efficiency, we need to know the enthalpies at all other states:

State 1: $P_1 = 10 \text{ kPa}$, Sat. liquid $\Big\}$ $h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81 \text{ kJ/kg}$; $v_1 = v_{f\,@\,10\,\text{kPa}} = 0.00101 \text{ m}^3/\text{kg}$

State 2: $P_2 = 15 \text{ MPa}$

$s_2 = s_1$

$$w_{\text{pump,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg}) \times [(15{,}000 - 10)\text{kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= 15.14 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump,in}} = (191.81 + 15.14) \text{ kJ/kg} = 206.95 \text{ kJ/kg}$$

State 3: $P_3 = 15 \text{ MPa}$, $T_3 = 600°\text{C}$ $\Big\}$ $h_3 = 3583.1 \text{ kJ/kg}$; $s_3 = 6.6796 \text{ kJ/kg}\cdot\text{K}$

State 4: $P_4 = 4 \text{ MPa}$, $s_4 = s_3$ $\Big\}$ $h_4 = 3155.0 \text{ kJ/kg}$; $(T_4 = 375.5°\text{C})$

Thus

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4)$$

$$= (3583.1 - 206.95) \text{ kJ/kg} + (3674.9 - 3155.0) \text{ kJ/kg}$$

$$= 3896.1 \text{ kJ/kg}$$

$$q_{\text{out}} = h_6 - h_1 = (2335.1 - 191.81) \text{ kJ/kg}$$

$$= 2143.3 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2143.3 \text{ kJ/kg}}{3896.1 \text{ kJ/kg}} = \mathbf{0.450 \text{ or } 45.0\%}$$

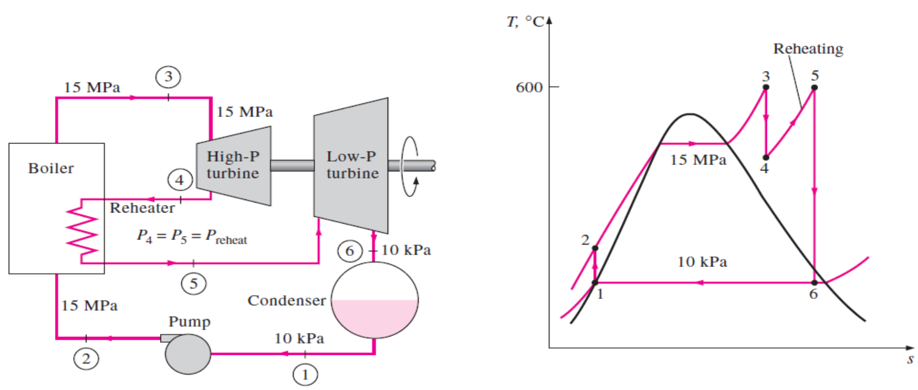


FIGURE 10–13
Schematic and *T-s* diagram for Example 10–4.

10–6 ▪ THE IDEAL REGENERATIVE RANKINE CYCLE

A careful examination of the *T-s* diagram of the Rankine cycle redrawn in Fig. 10–14 reveals that heat is transferred to the working fluid during process 2-2′ at a relatively low temperature. This lowers the average heat-addition temperature and thus the cycle efficiency.

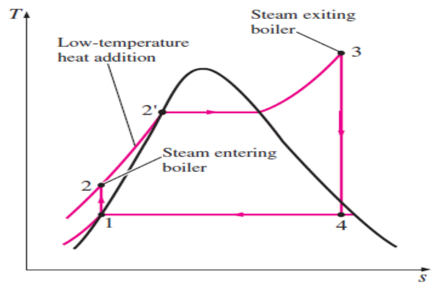


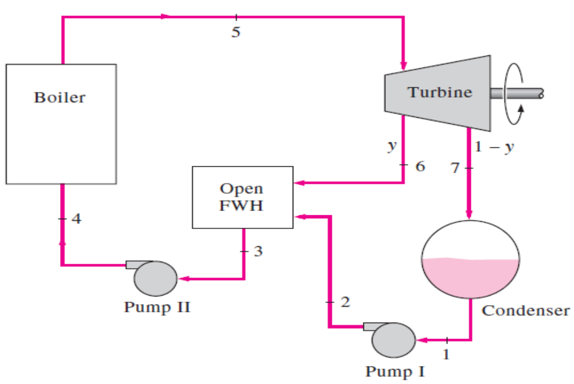
FIGURE 10–14
The first part of the heat-addition process in the boiler takes place at relatively low temperatures.

$$q_{in} = h_5 - h_4 \quad (10\text{–}14)$$

$$q_{out} = (1 - y)(h_7 - h_1) \quad (10\text{–}15)$$

$$w_{turb,out} = (h_5 - h_6) + (1 - y)(h_6 - h_7) \quad (10\text{–}16)$$

$$w_{pump,in} = (1 - y)w_{pump\ I,in} + w_{pump\ II,in} \quad (10\text{–}17)$$



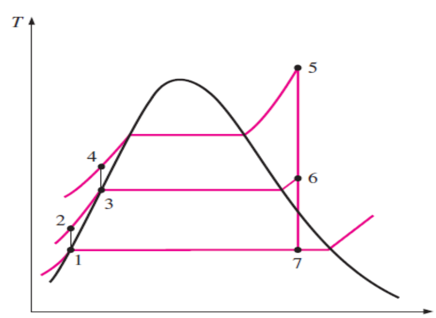


FIGURE 10–15
The ideal regenerative Rankine cycle with an open feedwater heater.

where

$$y = \dot{m}_6/\dot{m}_5 \quad \text{(fraction of steam extracted)}$$

$$w_{pump\ I,in} = v_1(P_2 - P_1)$$

$$w_{pump\ II,in} = v_3(P_4 - P_3)$$

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EXAMPLE 10–5 The Ideal Regenerative Rankine Cycle

Consider a steam power plant operating on the ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa.

Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open feedwater heater. Determine the fraction of steam extracted from the turbine and the thermal efficiency of the cycle.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–18. We note that the power plant operates on the ideal regenerative Rankine cycle. Therefore, the pumps and the turbines are isentropic; there are no pressure drops in the boiler, condenser, and feedwater heater; and steam leaves the condenser and the feedwater heater as saturated liquid. First, we determine the enthalpies at various states:

State 1: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \quad \begin{array}{l} h_1 = h_{f\,@\,10\text{ kPa}} = 191.81 \text{ kJ/kg} \\ v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg} \end{array}$

State 2: $P_2 = 1.2 \text{ MPa}$

$$s_2 = s_1$$

$$w_{\text{pump I,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3\text{/kg})[(1200 - 10) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= 1.20 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump I,in}} = (191.81 + 1.20) \text{ kJ/kg} = 193.01 \text{ kJ/kg}$$

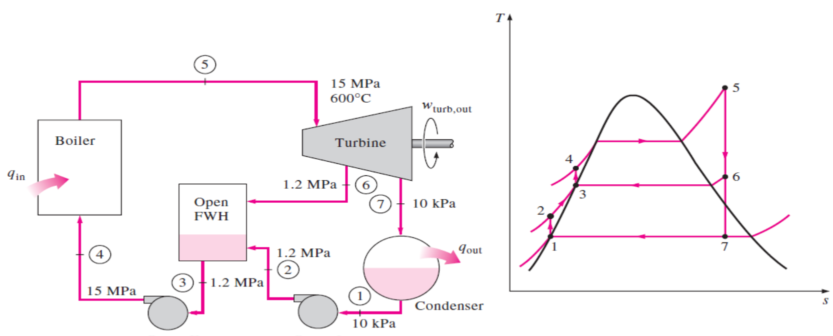


FIGURE 10–18
Schematic and *T-s* diagram for Example 10–5.

State 3: $\left.\begin{array}{l} P_3 = 1.2\text{ MPa} \\ \text{Sat. liquid} \end{array}\right\}$ $\begin{array}{l} v_3 = v_{f\,@\,1.2\text{ MPa}} = 0.001138\text{ m}^3/\text{kg} \\ h_3 = h_{f\,@\,1.2\text{ MPa}} = 798.33\text{ kJ/kg} \end{array}$

State 4: $P_4 = 15\text{ MPa}$

$$s_4 = s_3$$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3)$$

$$= (0.001138\text{ m}^3/\text{kg})[(15{,}000 - 1200)\text{ kPa}]\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 15.70\text{ kJ/kg}$$

$$h_4 = h_3 + w_{\text{pump II,in}} = (798.33 + 15.70)\text{ kJ/kg} = 814.03\text{ kJ/kg}$$

State 5: $\left.\begin{array}{l} P_5 = 15\text{ MPa} \\ T_5 = 600°\text{C} \end{array}\right\}$ $\begin{array}{l} h_5 = 3583.1\text{ kJ/kg} \\ s_5 = 6.6796\text{ kJ/kg}\cdot\text{K} \end{array}$

State 6: $\left.\begin{array}{l} P_6 = 1.2\text{ MPa} \\ s_6 = s_5 \end{array}\right\}$ $\begin{array}{l} h_6 = 2860.2\text{ kJ/kg} \\ (T_6 = 218.4°\text{C}) \end{array}$

State 7: $P_7 = 10\text{ kPa}$

$$s_7 = s_5 \qquad x_7 = \frac{s_7 - s_f}{s_{fg}} = \frac{6.6796 - 0.6492}{7.4996} = 0.8041$$

$$h_7 = h_f + x_7 h_{fg} = 191.81 + 0.8041(2392.1) = 2115.3\text{ kJ/kg}$$

The energy analysis of open feedwater heaters is identical to the energy analysis of mixing chambers. The feedwater heaters are generally well insulated ($\dot{Q} = 0$), and they do not involve any work interactions ($\dot{W} = 0$). By neglecting the kinetic and potential energies of the streams, the energy balance reduces for a feedwater heater to

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow \sum_{\text{in}} \dot{m}h = \sum_{\text{out}} \dot{m}h$$

or

$$yh_6 + (1 - y)h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($=\dot{m}_6/\dot{m}_5$). Solving for y and substituting the enthalpy values, we find

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{798.33 - 193.01}{2860.2 - 193.01} = \mathbf{0.2270}$$

Thus,

$$q_{\text{in}} = h_5 - h_4 = (3583.1 - 814.03)\text{ kJ/kg} = 2769.1\text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81)\text{ kJ/kg}$$

$$= 1486.9\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1486.9\text{ kJ/kg}}{2769.1\text{ kJ/kg}} = \mathbf{0.463\ or\ 46.3\%}$$

EXAMPLE 10–6 The Ideal Reheat–Regenerative Rankine Cycle

Consider a steam power plant that operates on an ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam is extracted from the turbine at 4 MPa for the closed feedwater heater, and the remaining steam is reheated at the same pressure to 600°C. The extracted steam is completely condensed in the heater and is pumped to 15 MPa before it mixes with the feedwater at the same pressure. Steam for the open feedwater heater is extracted from the low-pressure turbine at a pressure of 0.5 MPa. Determine the fractions of steam extracted from the turbine as well as the thermal efficiency of the cycle.

Analysis The schematic of the power plant and the *T-s* diagram of the cycle are shown in Fig. 10–19. The power plant operates on the ideal reheat–regenerative Rankine cycle and thus the pumps and the turbines are isentropic; there are no pressure drops in the boiler, reheater, condenser, and feedwater heaters; and steam leaves the condenser and the feedwater heaters as saturated liquid.

The enthalpies at the various states and the pump work per unit mass of fluid flowing through them are

$$h_1 = 191.81 \text{ kJ/kg} \qquad h_9 = 3155.0 \text{ kJ/kg}$$
$$h_2 = 192.30 \text{ kJ/kg} \qquad h_{10} = 3155.0 \text{ kJ/kg}$$
$$h_3 = 640.09 \text{ kJ/kg} \qquad h_{11} = 3674.9 \text{ kJ/kg}$$
$$h_4 = 643.92 \text{ kJ/kg} \qquad h_{12} = 3014.8 \text{ kJ/kg}$$
$$h_5 = 1087.4 \text{ kJ/kg} \qquad h_{13} = 2335.7 \text{ kJ/kg}$$
$$h_6 = 1087.4 \text{ kJ/kg} \qquad w_{\text{pump I,in}} = 0.49 \text{ kJ/kg}$$
$$h_7 = 1101.2 \text{ kJ/kg} \qquad w_{\text{pump II,in}} = 3.83 \text{ kJ/kg}$$
$$h_8 = 1089.8 \text{ kJ/kg} \qquad w_{\text{pump III,in}} = 13.77 \text{ kJ/kg}$$

The fractions of steam extracted are determined from the mass and energy balances of the feedwater heaters:

Closed feedwater heater:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$yh_{10} + (1 - y)h_4 = (1 - y)h_5 + yh_6$$

$$y = \frac{h_5 - h_4}{(h_{10} - h_6) + (h_5 - h_4)} = \frac{1087.4 - 643.92}{(3155.0 - 1087.4) + (1087.4 - 643.92)} = 0.1766$$

Open feedwater heater:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$zh_{12} + (1 - y - z)h_2 = (1 - y)h_3$$

$$z = \frac{(1 - y)(h_3 - h_2)}{h_{12} - h_2} = \frac{(1 - 0.1766)(640.09 - 192.30)}{3014.8 - 192.30} = 0.1306$$

The enthalpy at state 8 is determined by applying the mass and energy equations to the mixing chamber, which is assumed to be insulated:

$$\dot{E}_{in} = \dot{E}_{out}$$

$$(1)h_8 = (1 - y)h_5 + yh_7$$

$$h_8 = (1 - 0.1766)(1087.4)\ \text{kJ/kg} + 0.1766(1101.2)\ \text{kJ/kg}$$

$$= 1089.8\ \text{kJ/kg}$$

Thus,

$$q_{in} = (h_9 - h_8) + (1 - y)(h_{11} - h_{10})$$
$$= (3583.1 - 1089.8)\ \text{kJ/kg} + (1 - 0.1766)(3674.9 - 3155.0)\ \text{kJ/kg}$$
$$= 2921.4\ \text{kJ/kg}$$
$$q_{out} = (1 - y - z)(h_{13} - h_1)$$
$$= (1 - 0.1766 - 0.1306)(2335.7 - 191.81)\ \text{kJ/kg}$$
$$= 1485.3\ \text{kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1485.3\ \text{kJ/kg}}{2921.4\ \text{kJ/kg}} = \mathbf{0.492\ or\ 49.2\%}$$

Discussion This problem was worked out in Example 10–4 for the same pressure and temperature limits with reheat but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 45.0 to 49.2 percent as a result of regeneration.

The thermal efficiency of this cycle could also be determined from

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{w_{turb,out} - w_{pump,in}}{q_{in}}$$

where

$$w_{turb,out} = (h_9 - h_{10}) + (1 - y)(h_{11} - h_{12}) + (1 - y - z)(h_{12} - h_{13})$$
$$w_{pump,in} = (1 - y - z)w_{pump\ I,in} + (1 - y)w_{pump\ II,in} + (y)w_{pump\ III,in}$$

Also, if we assume that the feedwater leaves the closed FWH as a saturated liquid at 15 MPa (and thus at $T_5 = 342°C$ and $h_5 = 1610.3$ kJ/kg), it can be shown that the thermal efficiency would be 50.6.

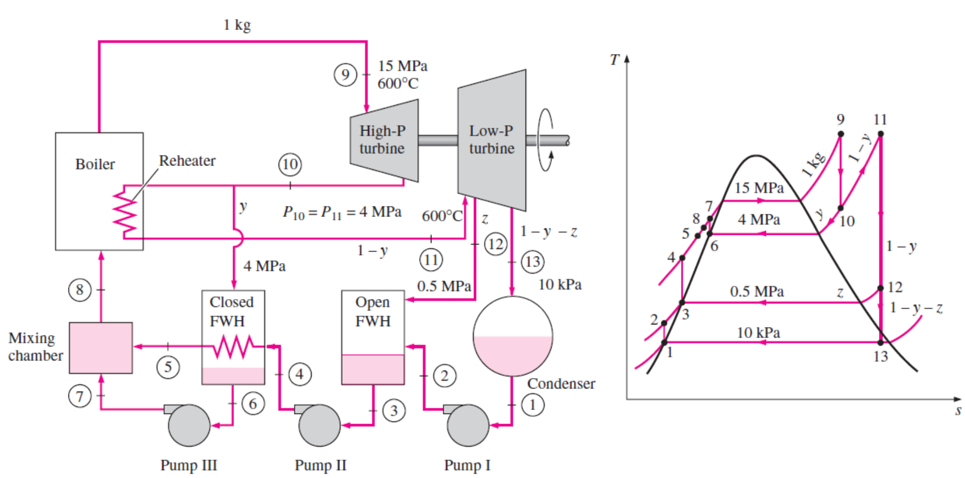
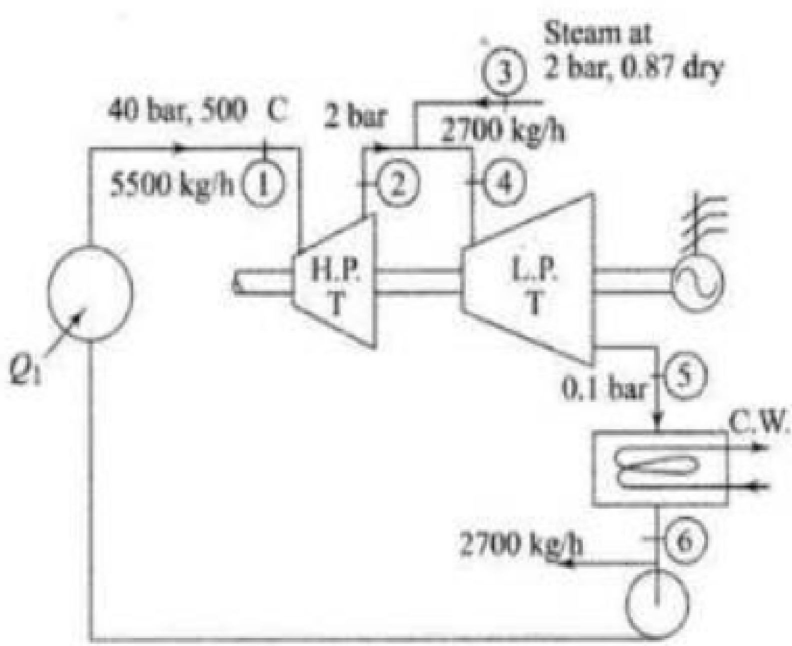


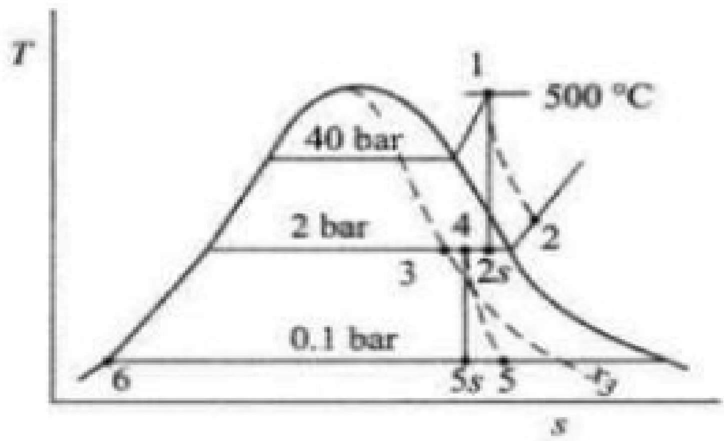
FIGURE 10–19
Schematic and *T-s* diagram for Example 10–6.

Steam Power – Problem 1

Steam at 40 bar, 500°C flowing at the rate of 5500 kg/h expands in a h.p. turbine to 2 bar with an isentropic efficiency of 83%. A continuous supply of steam at 2 bar, 0.87 quality and a flow rate of 2700 kg/h is available from a geothermal energy source. This steam is mixed adiabatically with the h.p. turbine exhaust steam and the combined flow then expands in a l.p. turbine to 0.1 bar with an isentropic efficiency of 78%. Determine the power output and the thermal efficiency of the plant. Assume that 5500 kg/h of steam is generated in the boiler at 40 bar, 500°C from the saturated feedwater at 0.1 bar.

Had the geothermal steam not been added what would have been the power output and efficiency of the plant? Neglect pump work.





SOLUTION:

With reference to the T-s diagram

h_1= 3445.3 KJ/kg

s_1 = 7.0901 KJ/kg K

$$s_1 = 7.0901 \text{ KJ/kg K} = 1.5301 + x_{2s}(5.5970)$$

$$x\ 2s = \frac{5.5600}{5.5970} = 0.9934$$

$$h_{2s} = 504.7 + 0.9934(2201.9) = 2692.04 \text{ KJ/kg K}$$

$$h_1 - h_2 = 0.83(3445.3 - 2692.04) = 625.21 \text{ KJ/kg}$$

$$h_2 = 3445.3 - 625.21 = 2820.09 \text{ KJ/kg}$$

$$h_3 = 504.7 + 0.87(2201.9) = 2420.4 \text{ KJ/kg}$$

$$2700h_3 + 5500h_2 = (2700 + 5500)h_4$$

$$h\ 4 = \frac{2700(2420.4) + 5500(2820.09)}{8200} = 796.96 + 1891.52$$

$$h_4 = 2688.48 \text{ KJ/kg}$$

$$h_4 = 504.7 + x_4(2201.9) = 2688.48$$

$$x_4 = \frac{2183.78}{2201.9} = 0.9917$$

$$s_4 = 1.5301 + 0.9917(5.5970) = 7.0806 \text{ KJ/kg K}$$

$$s_4 = s_{5s} = 0.6493 + x_{5s}(7.5009)$$

$$x_{5s} = 0.8574$$

$$h_{5s} = 191.84 + 0.8574(2392.8) = 2243.44 \text{ KJ/kg}$$

$$h_4 - h_5 = 0.78(2688.48 - 2243.44) = 347.1 \text{ KJ/kg}$$

$$h_6 = 191.83 \text{ KJ/kg}$$

$$W = 5500(h_1 - h_2) + 8200(h_4 - h_5)$$

$$W = 5500(625.21) + 8200(347.1)$$

$$\mathbf{W = 6\ 284\ 875\ KJ/h = 1745.8\ KW}$$

$$Q_1 = 5500(h_1 - h_6)$$

$$Q_1 = 5500(3445.3 - 191.8) \times 1/3600 = 4970.63 \text{ KW}$$

$$\eta_{cycle} = \frac{1745.8}{4970.63} = 0.353 = 35\%$$

Without geothermal heat supply:

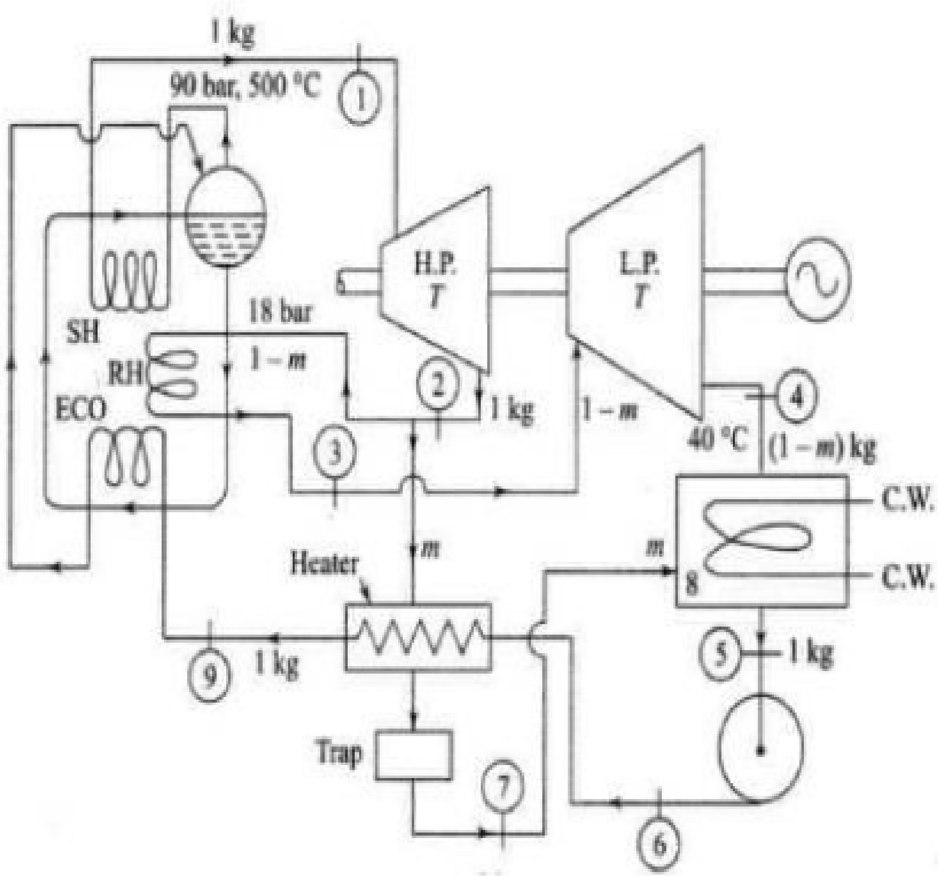
$$\mathbf{W_T} = 5500(h_1 - h_2) = \mathbf{955.18\ KW}$$

$$Q_1 = 5500(h_1 - h_6) = 4970.63 \text{ KW}$$

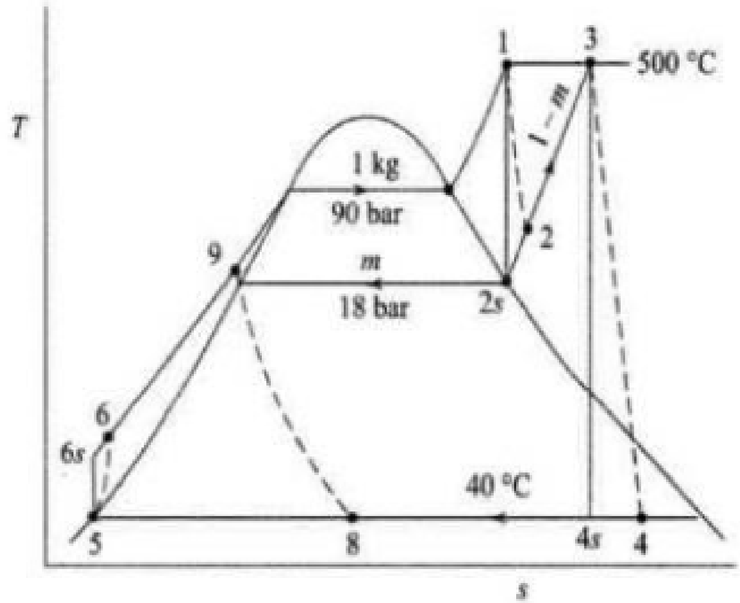
$$\eta_{cycle} = \frac{955.18}{4970.63} = 0.1922 = 19.22\%$$

Steam Power – Problem 2

A steam power plant with inlet steam to the h.p. turbine at 90 bar and 500°C, and condensation at 40°C produces 500 MW. It has one stage of reheat optimally placed which raises the steam temperature back to 500°C. One closed feedwater heater with drains cascaded back to the condenser receives bled steam at the reheat pressure, and the remaining steam is reheated and then expanded in the l.p. turbine. The h.p. and l.p. turbines have isentropic efficiencies of 92% and 90% respectively. The isentropic efficiency of the pump is 75%. Calculate (a) the mass flow rate of steam at turbine inlet in kg/s, (b) the cycle efficiency, and (c) the cycle work ratio. Use TTD = -1.6°C.



to



SOLUTION:

The flow and T-s diagram is shown. The optimum reheat pressure is taken to be 20% of the boiler pressure, which becomes 0.2 × 90 = 18 bar.

Now, h_1 = 3386.1 KJ/kg, s_1 = 6.6576 KJ/kg K = s_{2s}, h_{2s} = 2915 KJ/kg, h_3 = 3469.8 KJ/kg.

$$s_3 = 7.4825 = s_{4s} = s_f + x_{4s} s_{fg}$$

$$x_{4s} = \frac{7.4825 - 0.5725}{7.6845} = 0.8992$$

$$h_{4s} = 167.57 + 0.8992(2406.7) = 2331.7 \text{ KJ/kg}$$

$$h_5 = 167.57 \text{ KJ/kg}, \; h_7 = 883.42 \text{ KJ/kg}$$

$$W_{ps} = \int v\, dp = 0.001008 \times 90 \times 10 = 9.072 \text{ KJ/kg}$$

$$h_{6s} = 176.64 \text{ KJ/kg}$$

$$h_1 - h_2 = 0.92(3386 - 2915) = 433.3 \text{ KJ/kg, or}$$

$$h_2 = 3386.1 - 433.3 = 2952.8 \text{ KJ/kg}$$

$$h_3 - h_4 = 0.9(3469.8 - 2331.7) = 1024.29 \text{ KJ/kg}$$

$$h_4 = 3469.8 - 1024 = 2445.5 \text{ KJ/kg}$$

$$W_P = h_6 - h_5 = \frac{9.072}{0.75} = 12.1 \text{ KJ/kg} \quad \text{, or}$$

$$h_6 = 167.57 + 12.1 = 179.67 \text{ KJ/kg}$$

$$t_{sat} \text{ at 18 bar} = 207.15°C \qquad \therefore t_9 = 207.15 + 1.6 = 208.75°C$$

$$h_9 = 875 \text{ KJ/kg}$$

$$1(h_9 - h_6) = m(h_2 - h_7), \qquad m = \frac{875 - 179.67}{2952.8 - 883.4} = 0.336 \text{ kg}$$

$$W_T = (h_1 - h_2) + (1 - m)(h_3 - h_4) = 433.3 + 0.664(1024.3) = 1113.435 \text{ KJ/kg}$$

$$W_{net} = W_T - W_P = 1113.435 - 12.1 = 1101.335 \text{ KJ/kg}$$

$$w\ s = \frac{500 \times 103}{1101.335} = 454 \text{ kg/s} \qquad \text{(a)}$$

$$Q_1 = h_1 - h_9 + (1 - m)(h_3 - h_2)$$

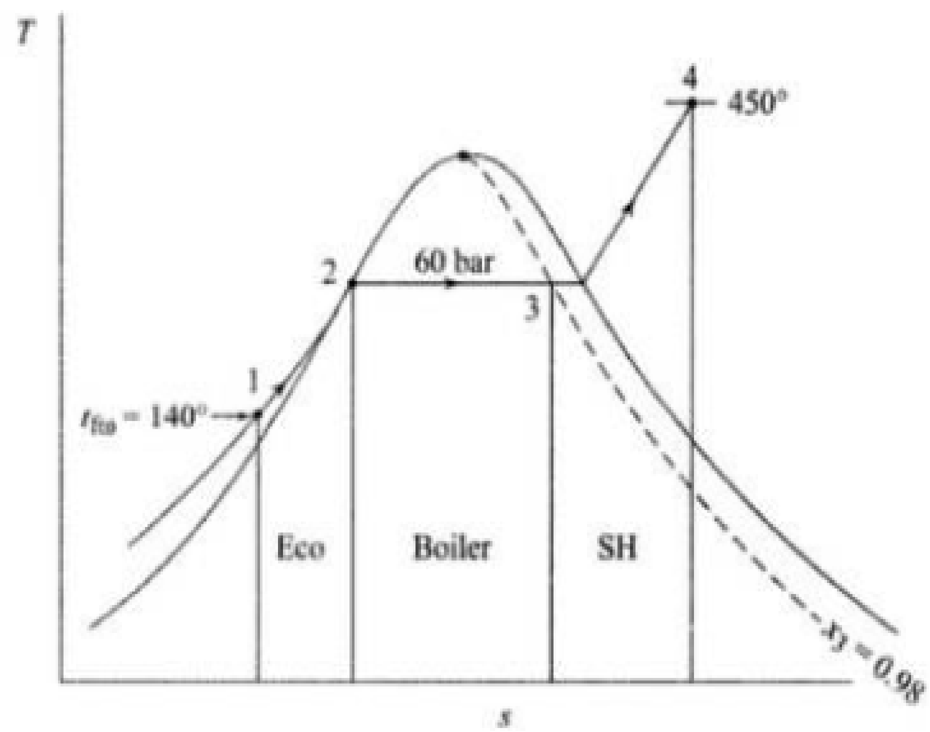
$$= 3386.1 - 875 + 0.664(3469.8 - 2952.8) = 2854.3 \text{ KJ/kg}$$

$$\eta cycle = \frac{1101.335}{2854.3} = 0.3858, \text{ or } 38.58\% \qquad \text{(b)}$$

$$\text{work ratio} = \frac{W\,net}{W\,T} = \frac{1101.335}{1113.435} = 0.989 \qquad \text{(C)}$$

Steam Power – Problem 3

A steam generator comprises a boiler, superheater, an economizer and an air preheater. The feedwater enters the economizer at 140°C and leaves as saturated liquid. Air is preheated from a temperature of 25°C to 250°C. Steam leaves the boiler drum at 60 bar, 0.98 dry and leaves the superheater at 450°C. When using coal with a calorific value of 25.2 MJ/kg, the rate of evaporation is 8.5 kg steam per kg coal and the air fuel ratio is 15:1 by mass. Neglecting heat losses and pressure drops, estimate the heat transfer per kg fuel in each component and the efficiency of the steam generator. What are the percentage of the total heat absorption taking place in the economizer, boiler and the superheater, respectively? Assume c_p of air and water as 1.005 and 4.2 kJ/kg K, respectively.



With reference to figure above

$h_1 = 140 \times 4.2 = 588$ kJ/kg

$h_f = 1213.35$ kJ/kg, $h_{fg} = 1571.0$ kJ/kg(at 60 bar)

$h_4 = 3301.8$ kJ/kg

$h_3 = 1213.35 + 0.98(1571.0) = 2752.93$ kJ/kg

$$e_{St.gen} = \frac{w_s(h_4 - h_1)}{w_s \times C.V} = \frac{8.5 \times 2713.8}{25.2 \times 1000}$$

$$e_{St.gen} = 0.9154 \quad \text{or} \quad 91.54\%$$

Heat transfer in the economizer

$$= \frac{w_s(h_2 - h_1)}{w_f} = 8.5(625.35 \times 10^{-3}) = 5.3155 \text{ MJ/kg}$$

Heat transfer in the boiler

$$= \frac{w_s(h_3 - h_2)}{w_f} = 8.5(1539.58 \times 10^{-3}) = 13.086 \text{ MJ/kg}$$

Heat transfer in the superheater

$$= \frac{w_s(h_4 - h_3)}{w_f} = 8.5(548.87 \times 10^{-3}) = 4.665 \text{ MJ/kg}$$

Heat transfer in the air preheater

$$= \frac{w_a c_{pa}(t_2 - t_1)}{w_f} = 15(1.005)(250 - 25) \times 10^{-3} = 3.392 \text{ MJ/kg}$$

Percentage of total heat absorbed in the economizer

$$= \frac{h_2 - h_1}{h_4 - h_1} \times 100 = \frac{625.35}{2713.8} \times 100 = 23.04\ \%$$

Percentage of total heat absorbed in the boiler